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USAAVLABS TECHNICAL REPORT 67-46

DESIGN STUDY OF HEAVY LIFT HELICOPTER EXTERNAL LOAD HANDLING SYSTEM

By

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November 1967

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-467(T)
SIKORSKY AIRCRAFT
DIVISION OF UNITED AIRCRAFT CORPORATION
STRATFORD, CONNECTICUT

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U S ARMY MATERIEL COMMAND

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The basis for this design study was obtained from previous investigations of problems associated with the mechanics of cargo handling by aerial-crane-type aircraft (USAAVLABS Technical Report 66-63). This report is one of three contract studies of the same problem with varying technical approaches. The conclusions drawn by this contractor are based on sound analytical techniques. They are particularly appropriate to a single-rotor, aerial-crane-type configuration. In this context, this command concurs in general with these findings. The preliminary designs developed by the contractor are complete, accurate, and in sufficient detail to provide a basis for component development programs.

Future work anticipated by this activity relative to this area includes an analysis of the three preliminary contract designs so as to define an optimum system based on the best features of each. This may be followed by component development and test of critical items as appropriate and the detail design, fabrication, and test of an experimental system.

Task 1X130901D332
Contract DA 44-177-AMC-467(T)
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November 1967

DESIGN STUDY OF

HEAVY LIFT HELICOPTER

EXTERNAL LOAD HANDLING SYSTEM

Sikorsky Engineering Report 50441

By

Lester R. Burroughs
and
Harold E. Ralsten

Prepared by

Sikorsky Aircraft
Division of United Aircraft Corporation
Stratford, Connecticut

for

U.S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

SUMMARY

This report presents the results of a two-phase feasibility and preliminary design study of load suspension configurations capable of meeting the external cargo handling system requirements of a 40,000-pound-payload heavy lift helicopter.

In Phase I, Design Analysis, both separate function configurations (those that incorporate individual single- and multi-point hoists) and combined function configurations (multi-point hoists used to perform both single- and multi-point missions) have been investigated for two external load handling system arrangements: single- plus two-point suspension and single- plus four-point load suspension.

This phase was primarily concerned with the investigation of hoist types; the methods of power transmission (to the hoists); and the selection of mechanical, hydraulic, and electrical components. A comparative evaluation of 13 system arrangements was made on the basis of weight, power, reliability, in-flight safety, versatility, and productivity.

The single- plus two-point and single- plus four-point systems determined to meet the heavy lift requirement best were presented to USAAVLABS, and the latter system was recommended for the Phase II, Preliminary Design.

Upon receipt of approval, the preliminary design of a load suspension system incorporating a mechanically driven single-point hoist and four hydraulically driven multi-point hoists was initiated. This phase of the study included the preparation of preliminary (layouts) drawings, load and stress analysis of all major components, and a maintainability and reliability analysis, as well as the preparation of a component development plan.

The single- plus four-point system described herein weighs 4974 pounds for a capacity of 40,000 pounds. The system has been designed such that the hoists of both systems are readily removable when missions requiring minimum empty weight are to be undertaken. For single-point operation only (four-point hoists removed), the system weighs 2738 pounds; for four-point missions (single-point hoist removed) the system weighs 2704 pounds.

Both single- plus four-point systems have potential for growth to 50,000 pounds with a minimum of modification. This increased capacity can be realized for a total weight increase of 270 pounds for both systems.

A typical sequence of operations for both single- and four-point systems is outlined in Appendix IV.

FOREWORD

This report covers a two-phase evaluation of external cargo handling systems for a 40,000-pound-payload heavy lift helicopter. This project was conducted during the 10-month period from July 1, 1966, through April 28, 1967, for the U.S. Army Aviation Materiel Laboratories (USAAVLABS) under Contract DA 44-177-AMC-467(T). Pertinent data upon which portions of this study were based were provided by the following: Bergen Wire Rope Company; The Lycoming Division of AVCO; Eastern Rotorcraft Corporation; Vickers Incorporated, Division of Sperry Rand Corporation; Taylor Devices, Incorporated; and Holex, Incorporated.

USAAVLABS technical direction was provided by Mr. J. Vichness, Chief, Air Cargo Systems Branch.

The principal investigators for Sikorsky Aircraft were L. R. Burroughs, Assistant Supervisor, Mechanical Design and Development Section, and H. E. Ralsten, Supervisor, Mechanical Accessories Group. Also making significant contributions to this effort were A. Korzun, Design Engineer, J. Kish, Senior Design Analyst of the Mechanical Design Section, and R. Fidler, Design Analyst of the Hydraulics Section.

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SYMBOLS

a	Acceleration
A	Area, Strength Factor
B.L.	Buttock Line
C.G.	Center of Gravity
C_o	Basic Static Capacity of Bearing
CONVEN	Conventional
d_c	Cable Diameter
d_g	Gear Diameter
d_i	Inside Diameter
d_m	Mean Diameter
d_o	Outside Diameter
d_p	Pinion Diameter
D	Diameter
E	Modulus of Elasticity
E_d	Drum Modulus of Elasticity
E_L	Cable Longitudinal Modulus of Elasticity
E_t	Cable Transverse Modulus of Elasticity
f_b	Bending Stress
f_c	Compressive Stress
$f_{r\ max}$	Maximum Radial Stress
f_t	Torsional Shear Stress
$f_{t\ max}$	Maximum Tangential Stress
F	Gear Face Width
F_a	Axial Clutch Load

F_b	Allowable Bending Stress
F_c	Allowable Compressive Stress
F_{cy}	Compressive Yield Point
FH	Flight Hours
F_n	Normal Force
F.S.	Fuselage Station, Factor of Safety
F_{tu}	Ultimate Tensile Strength
g	Gravitational Constant
gpm	Gallons per Minute
H.L.H.	Heavy Lift Helicopter
HP	Horsepower
I	Moment of Inertia
J	Polar Moment of Inertia
K	Drum Pressure Constant
Kn	Knots
K_t	Stress Concentration Factor
l, L	Length
l_{drum}	Lead of Drum
l_{screw}	Lead of Screw
m	Mass
M_b	Bending Moment
M_{bv}	Vertical Bending Moment
M_{bH}	Horizontal Bending Moment
MF	Hydraulic Motor
MMH	Maintenance Man-Hours
M.S.	Margin of Safety

M_t	Torsional Moment
MTBUMA	Mean Time Between Unscheduled Maintenance Action
n	Number of Friction Surfaces in Brake or Clutch
n_1	Number of Layers of Cable
n.m.	Nautical Miles
N_ψ	Change in Directional Restoring Stability
OGE	Out of Ground Effect
p_c	Cable Pitch
psi	Pounds per Square Inch
P_a	Axial Load, Pressure on Clutch or Brake Plates
P_{allow}	Allowable Load
P_c	Cable Load
P_{cr}	Critical Buckling Load
P_e	External Pressure on Drum
PP	Hydraulic Pump
P_{limit}	Cable Limit Load
P_{ult}	Cable Ultimate Load
rpm	Revolutions per Minute
R	Bearing Reaction
RES	Reservoir
RR	Reduction Ratio
S	Distance
t	Thickness, Time
T	Torque, Tension
W	Weight
W.L.	Water Line

W_t	Tangential Tooth Load
X	Tooth Form Factor
Z	Section Modulus
Zero Mom	Zero Moment
α	Lead Angle, Angular Acceleration
γ	Cable Pressure Angle
δ Δ	Deflection
Σ	Summation
η	Efficiency
θ	Fore and Aft Cable Angle
ϕ	Side Cable Angle
ρ	Density
μ	Coefficient of Friction
ν	Poisson's Ratio
ω	Angular Velocity

INTRODUCTION

The primary purpose of the heavy lift helicopter is that of an aerial hoist and transporter for heavy loads including combat vehicles and other large, bulky items which cannot be lifted by other helicopters. Therefore, an initial consideration in the design of a 40,000-pound-payload heavy lift helicopter must necessarily be concerned with its external load handling winch and hoist system.

To provide good reliability and maintainability, adequate in-flight safety, and simple and accurate controls at a minimum weight, the cargo handling system must be designed concurrent with the helicopter airframe, particularly in those areas of interface. The basic airframe configurations used for this evaluation are those of the previous heavy lift transmission and rotor system studies conducted by Sikorsky Aircraft for USAAVLABS (References 3 and 4). The missions used for productivity analyses are also those of the previous studies.

PHASE I

DESIGN ANALYSIS

DISCUSSION

In this initial phase, both separate function configurations and combined function configurations will be investigated for two basic external load handling system arrangements: single-point plus two-point load suspension and single-point plus four-point load suspension.

Separate function configurations are those systems that incorporate individual single-point and multi-point hoists. Combined functions are those where the multi-point hoists are used to perform both the single and multi-point missions. The hoist systems to be evaluated will include those aircraft related components necessary for control, load attachment, suspension, hoisting, and isolation of the load from the aircraft.

The systems will be evaluated on the basis of power requirements, system efficiency, weight, reliability, safety, maintainability, cost, and technical confidence. At the completion of Phase I, Design Analysis, the single- plus two-point and single- plus four-point systems that best meet the cargo handling requirements of a 40,000-pound heavy lift helicopter will be selected and one of these will be recommended for preliminary design in Phase II.

BASIC DATA

DESIGN REQUIREMENTS

The heavy lift helicopter external load handling system evaluated herein has been designed to meet the requirements shown in Table I.

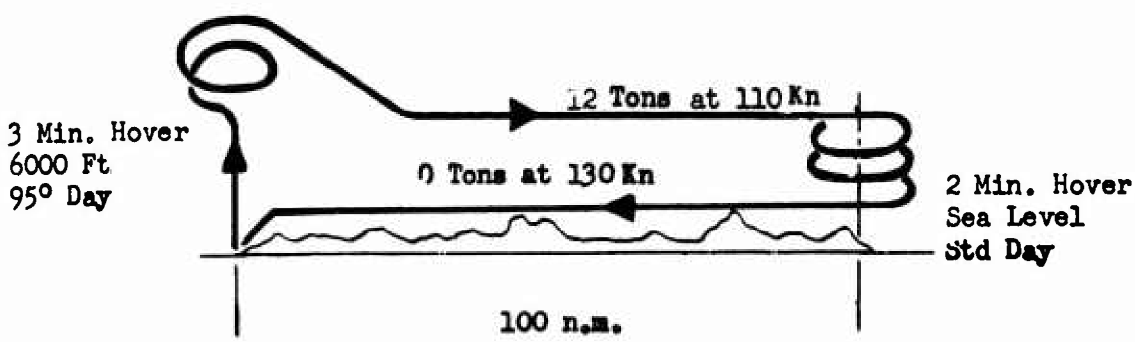
TABLE I
BASIC DESIGN DATA

	Single-Point	Two-Point (Per Hoist)	Four-Point (Per Hoist)
Load (lb)	40,000	23,100	11,550
Ultimate Load Factor	3.75	4.2	4.2
Usable Cable Length (ft)	150/80	50	50
Cable Angle - Static - Dynamic	$\pm 30^\circ$ $\pm 15^\circ$	$\pm 30^\circ$ $\pm 15^\circ$	$\pm 30^\circ$ $\pm 15^\circ$
Minimum Cable Speed (fpm)	60	30	30
Minimum Service Interval (cycles)		1200	
Minimum Retirement Interval (cycles)		3600	
System Weight Goal (lb)		4000	

MISSION REQUIREMENTS

As an aid in evaluating the various external cargo handling system configurations covered herein, the following mission spectra were assumed for the 40,000-pound-payload heavy lift helicopter. It has also been assumed that there is an equal frequency of occurence of each mission.

Transport Mission

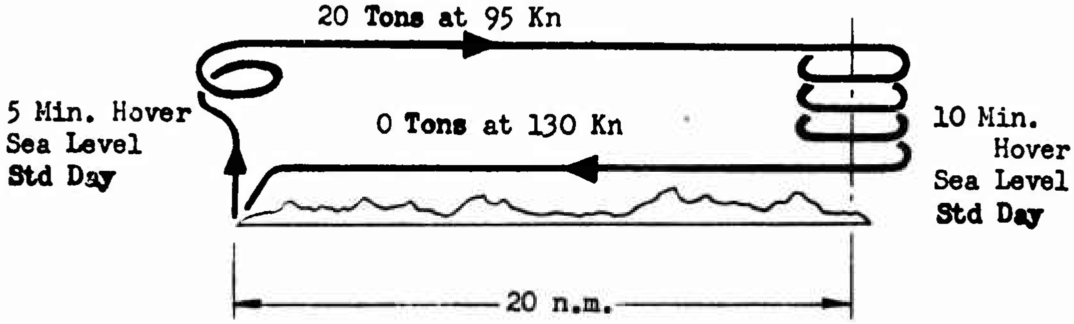


Transport Mission

Payload	12 tons (outbound)
Radius	100 n.m.
Vcruise	110 knots (12-ton payload)
Vcruise	130 knots (no payload)
Hovering Time	3 min. at takeoff (with 12-ton payload)
	2 min. at midpoint
Reserve Fuel	10% of initial fuel
Hovering Capability	6,000 ft, 95°F (OGE), takeoff gross
Mission Altitude	Sea level, standard atmosphere

Fuel Allowance for Start, Warm-up, and Takeoff
MIL-C-5011A

Heavy Lift Mission



Payload	20 tons (outbound)
Radius	20 n.m.
Vcruise	95 knots (20-ton payload)
Vcruise	130 knots (no payload)
Hovering Time	5 min. at takeoff
	10 min. at destination with payload
Reserve Fuel	10% of initial fuel
Hover Capability	Sea level, standard atmosphere

Fuel Allowance for Start, Warm-up, and Takeoff
MIL-C-5011A

AIRCRAFT DESCRIPTION

Figures 1 and 2, pages 5 and 7 describe the single- and tandem-rotor heavy lift helicopters, respectively, used as the aerial vehicles for the cargo handling configuration studies covered herein.

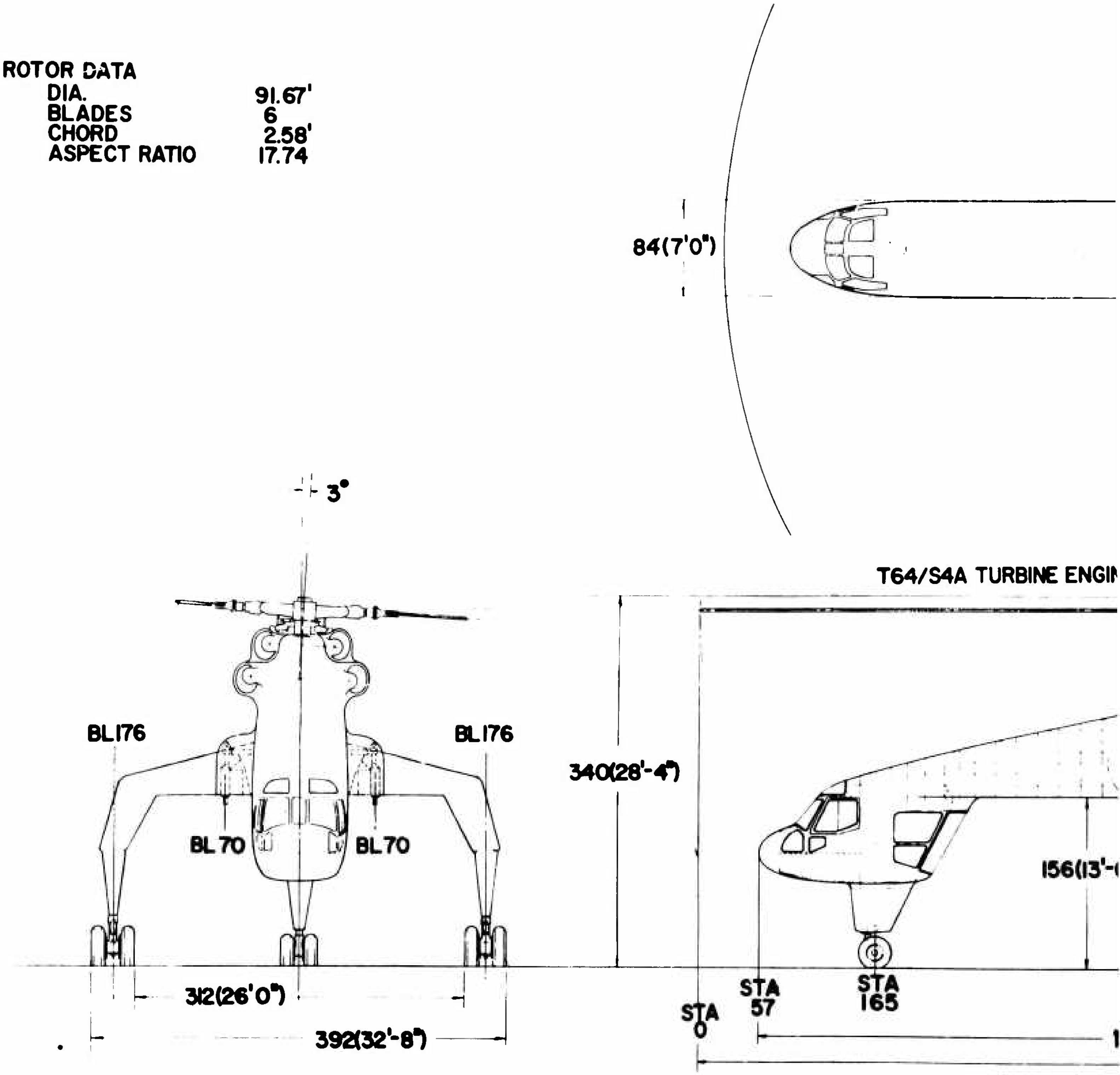
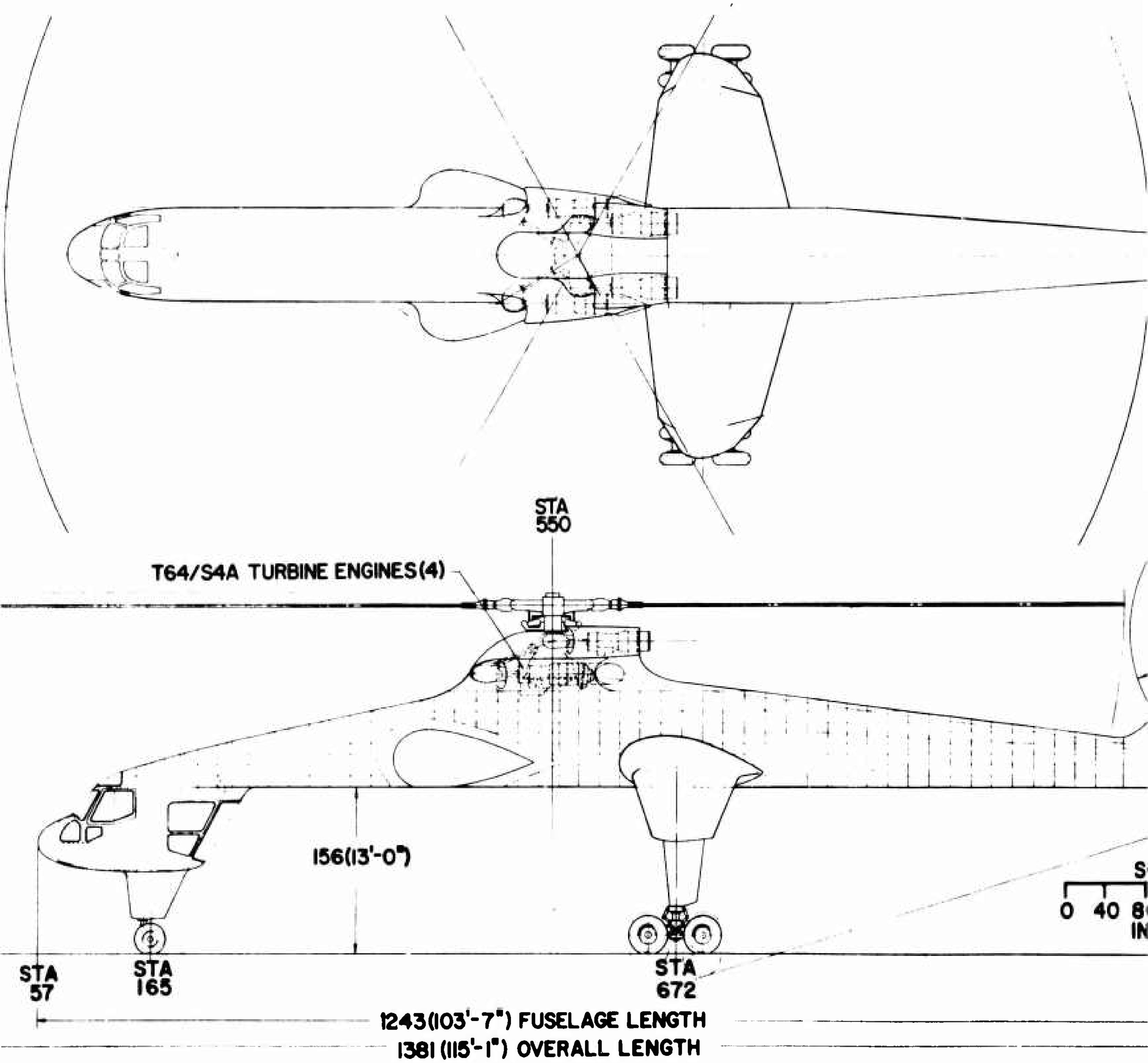
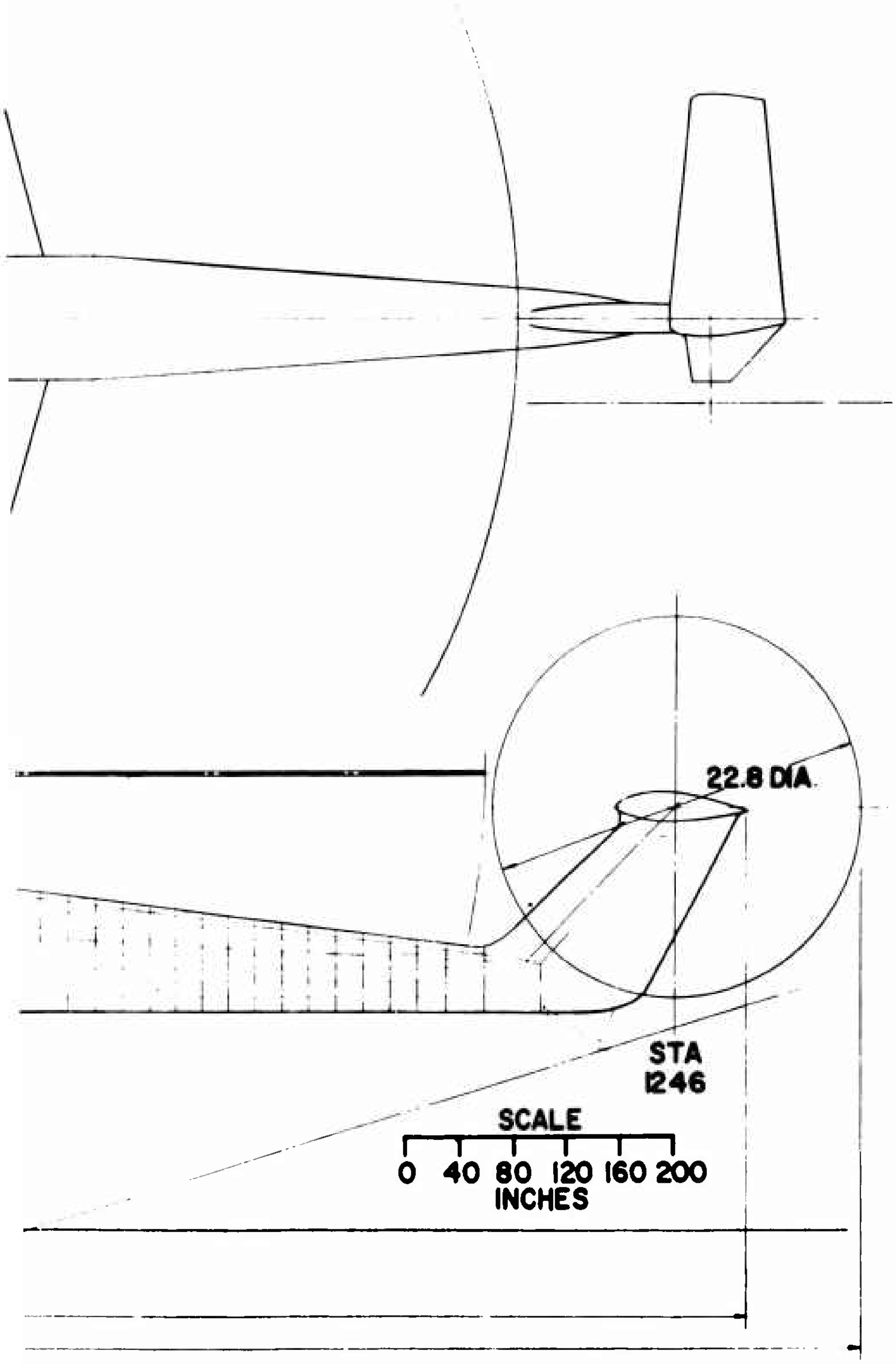


Figure 1. Single-Rotor H.L.H.

A,


STA 550
T64/S4A TURBINE ENGINES (4)
156 (13'-0")
0 40 8
IN
STA 57
STA 165
STA 672
1243 (103'-7") FUSELAGE LENGTH
1381 (115'-1") OVERALL LENGTH

B.



C.

ROTOR DATA

DIA.	70.58
BLADE CHORD	3.76'
NO. OF BLADES	3
BLADE AREA	795.00'
OVERLAP	23.30'
ASPECT RATIO	9.40

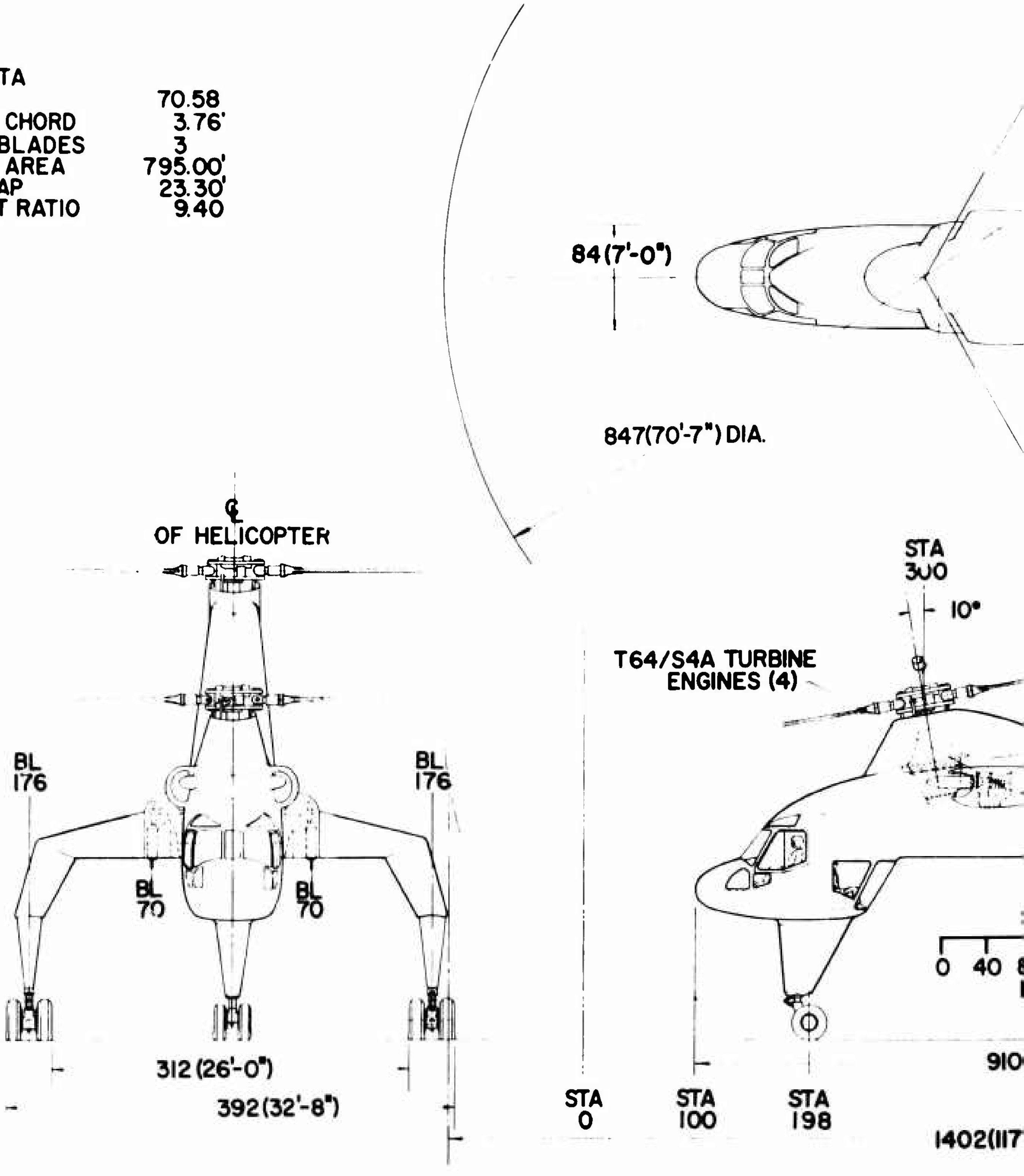
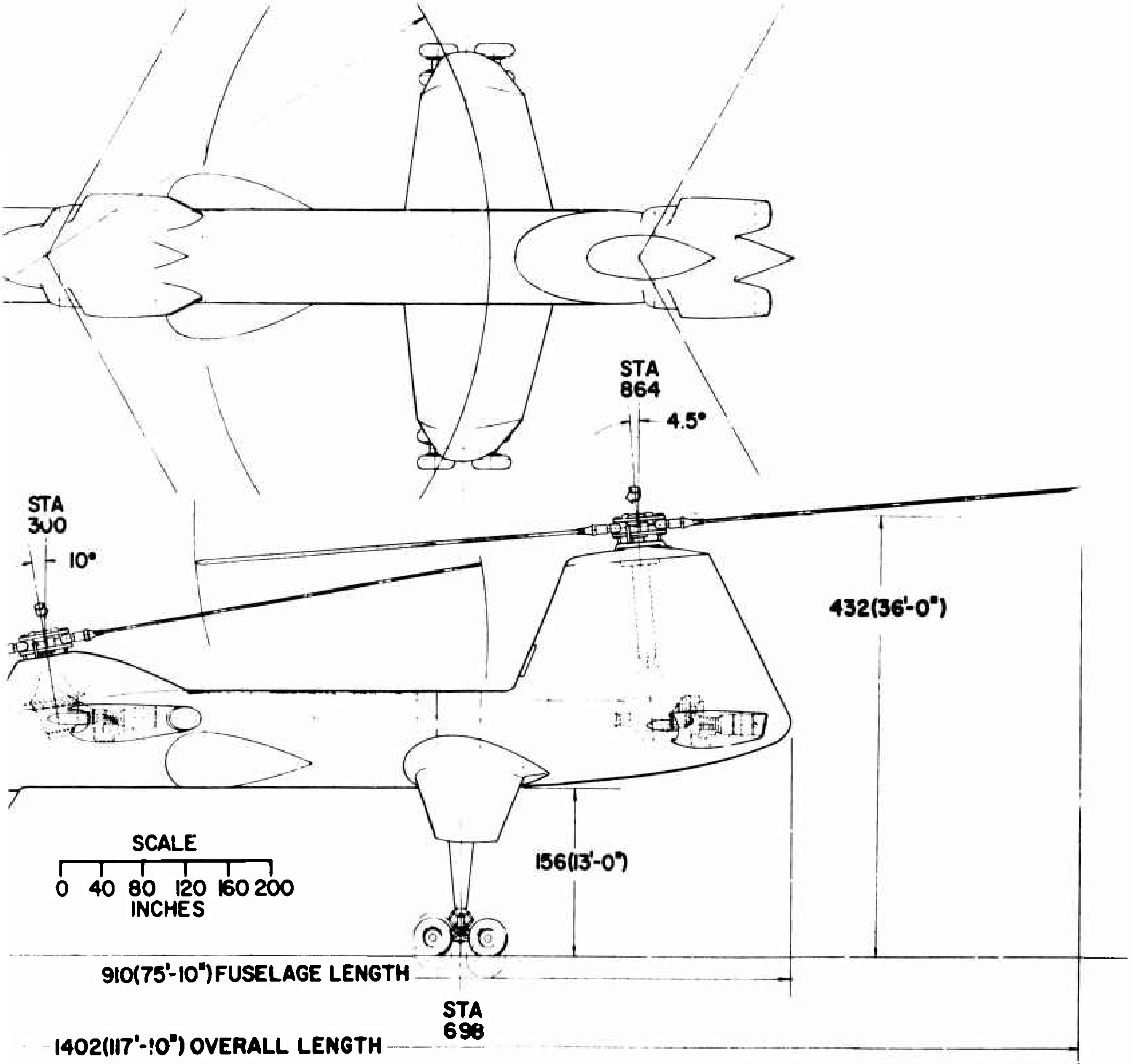


Figure 2. Tandem-Rotor H.L.H.

A.



STA 864
4.5°
STA 300
10°
432(36'-0")
SCALE
0 40 80 120 160 200
INCHES
156(13'-0")
910(75'-10")FUSELAGE LENGTH
STA 698
1402(117'-10") OVERALL LENGTH

R.

INVESTIGATION OF VEHICLES

To aid in the evaluation of various external load handling systems, it is necessary to survey the types of equipment to be carried. This information is basic, since it defines the aircraft-load interaction, the placement of hoists on the aircraft, and the means of load acquisition. Since the type of equipment to be carried was not specifically described in the contract, it was necessary to conduct a survey of the vehicles presently in use by the U.S. Army. A summary of the vehicles reviewed is contained in Appendix I.

Of the 94 vehicles listed, only 37 (items 49 thru 86 of Appendix I) were in the 15,000- to 40,000-pound weight class. Since vehicles below 15,000 pounds in weight are well within capabilities of other aircraft in the U.S. Army inventory (i.e., the CH-47A and CH-54A), they were not considered to have a major influence on the cargo handling system design. Table II shows average, maximum, and minimum dimensions of vehicles in the 15,000- to 40,000-pound weight range. These envelope dimensions were used as an aid in the placement of the multi-point hoists on the aircraft.

TABLE II
U.S. ARMY VEHICLES - LENGTH AND WIDTH DIMENSIONS,
15,000- to 40,000-POUND CLASS

	Minimum	Average	Maximum
Length, Inches	169	265.5	600
Width, Inches	92	106.5	145
Vehicle Designation	M-114	None	CH-47
Item Number*	49	-	56
Net Weight, Pounds	15,000	-	16,000

*Item number in Appendix I

A list of vehicles greater than 130 inches in height is shown in Table III. Three vehicles (the CH-47, the M-129 STLR Van, and the M-292 Truck Van) allow less that what is considered to be a practical clearance for any form of ground type pickup. The unloaded ground clearance of both single- and tandem-rotor heavy lift helicopters is 156 inches (Reference Figures 1 and 2, pages 5 and 7). Therefore, a clearance of less than 17 inches is considered inadequate due to the landing gear oleo compression during the hoisting of vehicles. This dictates hovering pickup for vehicles of this size.

TABLE III
U.S. ARMY VEHICLES WITH HEIGHTS OF 130 INCHES OR MORE,
15,000- to 40,000-POUND CLASS

Item Number*	Name	Height
50	M 313 STLR Van Equip.	134
51	M 220 Truck Van	131
54	M 109 Truck Van	130
56	CH-47 Helicopter	222
57	M 129 STLR Van	140
62	M 292 Truck Van Equip.	139

*Item number in Appendix I

It was impossible, within the scope and time frame of this study, to complete investigation of the aerodynamic properties of all 37 vehicles in the single- , two- , and four-point hoist suspension modes. Therefore, a representative number of vehicles, as shown in Table IV, were selected as typical and formed the basis for the hoist system configuration and aircraft-load interaction study.

TABLE IV
TYPICAL U.S. ARMY VEHICLES

Item Number*	Name	Length (Inches)	Width (Inches)	Height (Inches)	Weight (Pounds)
38	155 MM Howitzer	190	74	64	12,700
49	Personnel Carrier	169	92	80	15,276
83	5-Ton Wrecker Med.	310	97	103	33,320
86	Self-Propelled Mortar	221	129	109	43,200

*Item number in Appendix I

Part of the efficiency of the four-point system is derived from its ability to be hooked directly to fittings on many types of loads (see page 94). For this reason, more data on the size, location, and structural adequacy of pickup points on all of the 37 vehicles is needed to finalize the hoist locations.

While these data are not required if the two-point system is used (see page 91), it would be valuable if the size, location, and structural adequacy of the pickup points were known. The use of slings that can be attached directly to fittings on the vehicle will be simpler and more efficient than the development of methods for attachment of a standard nylon web sling to the underside of all the 37 vehicles.

HOIST SYSTEM AND COMPONENTS DESIGN

HOIST LOCATION AND TYPE

Introduction

The external cargo handling systems to be evaluated herein will incorporate provisions for lifting and carrying 40,000 pounds in both single- and multi-point modes. Systems that incorporate individual single- and multi-point hoists (separate function) and those that employ one system to perform both modes of operation (combined function) will be studied for both single- and tandem-rotor aircraft.

Hoist Location - Single-Rotor Aircraft

The single-point hoist for the single-rotor aircraft of Figure 1 is located directly under the main rotor at F.S. 550 to minimize the effect of load oscillations on aircraft stability. It is located in a well in the fuselage and will not extend below the airframe when the hook is in the full-up position. This will permit a cargo or personnel pod to be carried by the multi-point hoists without removal of the single-point hoists. The main gearbox support structure can be utilized to provide the required hoist mounting with a minimum increase in weight.

The two-point hoists are located on B.L. 0 at F.S. 406 and F.S. 694. The horizontal spacing of 288 inches was based on the survey of military vehicles in the 15,000- to 40,000-pound weight class. It allows most of the vehicles in this category to be lifted off the ground in the ground pickup mode. The hoists are located in wells in the fuselage so that they do not extend below the airframe when the hooks are in the full-up position. This permits the pod to be pulled up and locked to the fuselage. This relatively high hoist location also reduces the effect of lateral load oscillations on aircraft controllability, since the cable reaction point is quite close to the location of the center of gravity of the aircraft. Hoist well size requires the addition of approximately 40 pounds to the airframe structure. The selection of these locations for the two-point hoists may conflict with the airframe designers desirable location for fuel cells.

The four-point hoists are located on B.L. 70 and at F.S. 406 and F.S. 694. Horizontal and lateral spacing was selected to achieve compatibility during ground pickup with the widest variety of loads. The hoists are universally mounted on a davit type structure with suitable aerodynamic fairing, and they do not extend below the fuselage. This permits a pod to be pulled up and locked to the fuselage. As in the two-point system, this relatively high hoist location also reduces the effect of lateral load oscillations on aircraft controllability, since the cable reaction point is quite close to the location of the center of gravity of the aircraft. The universal-type mounting permits the hoists to be pivoted in order to reach attachment points on outsized vehicles without inducing heavy side loads on the hoist. Figure 3, page 13, shows the load attachment point space envelope and its variation with distance between hoist and the

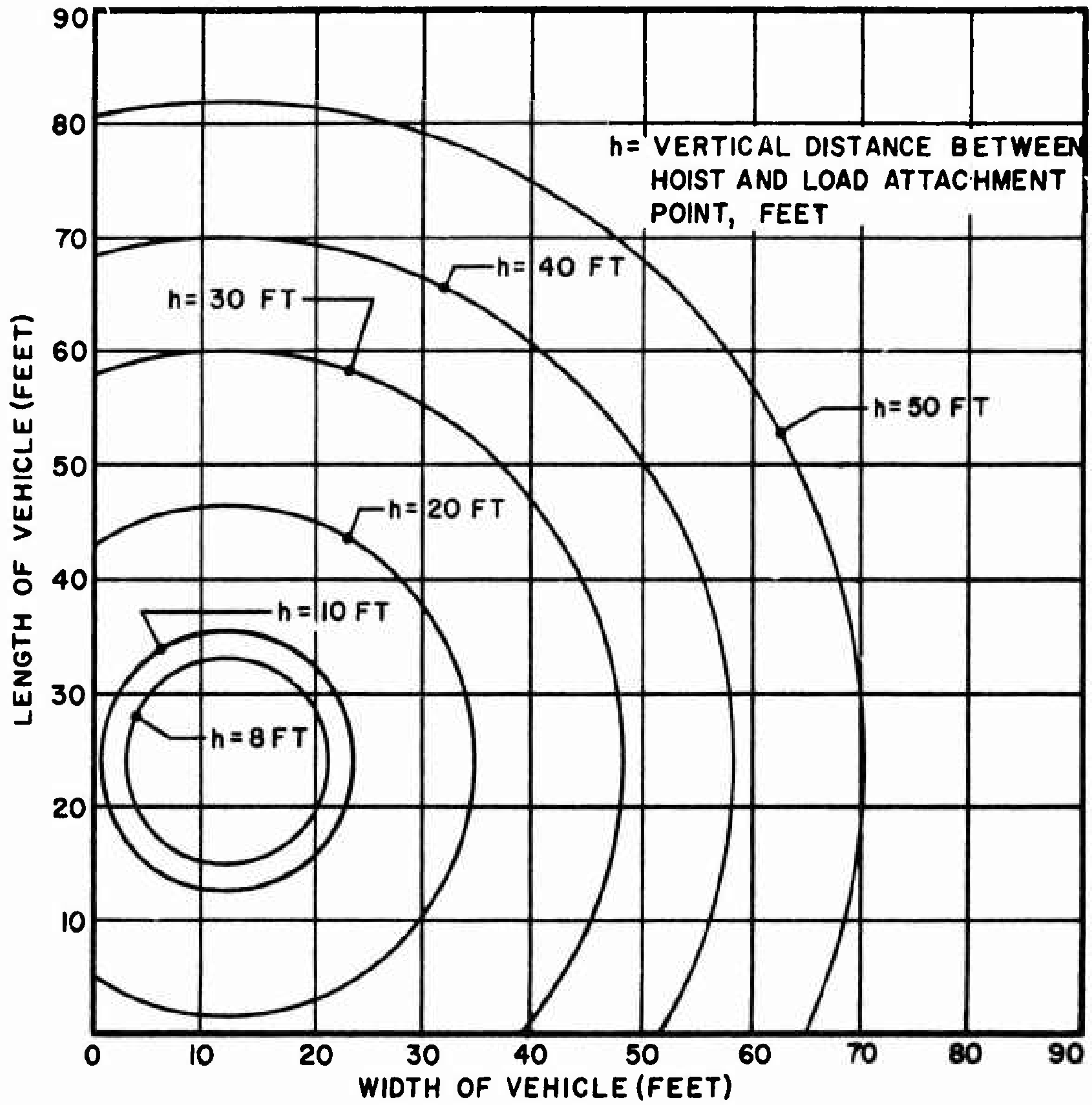


Figure 3. Four-Point Hoist Load Attachment Point Space Envelope.

Note: Area enclosed by circles gives physical dimensions of pickup points on loads that can be lifted without exceeding the permissible 60° cone angle for the four-point hoists at cable lengths specified.

ground with the hoist locations and cable angles selected. Page 22 contains an explanation of the cable angle requirement.

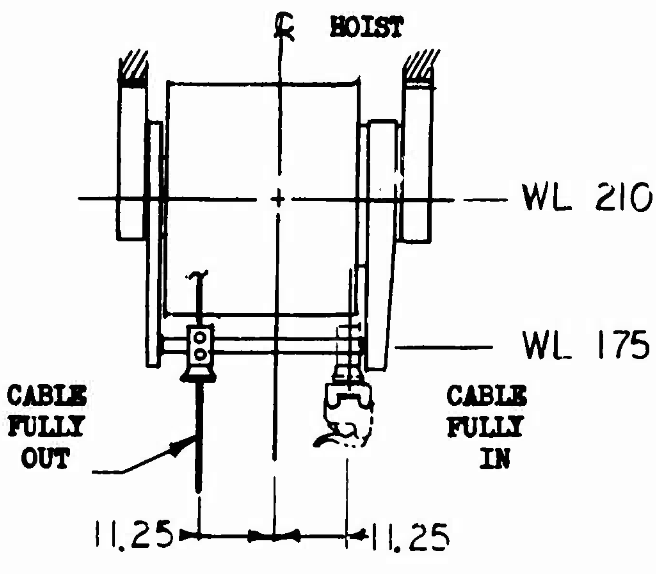
TABLE V
HOIST LOCATION* - SINGLE-ROTOR AIRCRAFT

	Single-Point Hoist	Two-Point Hoist	Four-Point Hoist
Fuselage Station	550	406 & 694	406 & 694
Buttock Line	11.25 right to 11.25 left**	0	70
Waterline:			
Longitudinal Cable Swing	210	200	200
Lateral Cable Swing	175	200	200

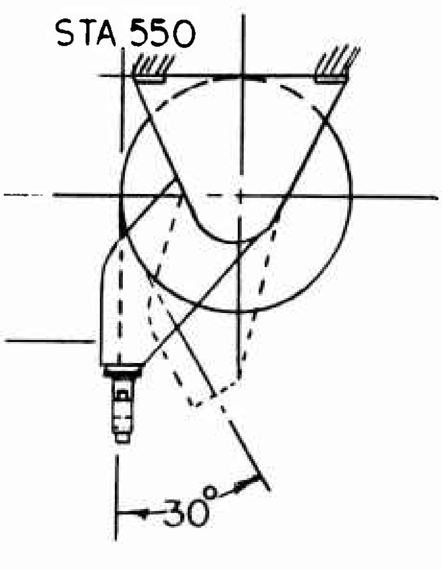
*Locations described refer to actual cable reaction point

**With 150 feet of cable

The actual cable reaction points given in Tables V and VI indicate a variation in the waterline location for lateral and longitudinal cable swing for the single-point hoist. This variation, as illustrated in the sketch below, is due to the basic design of the hoist.



FRONT VIEW



SIDE VIEW

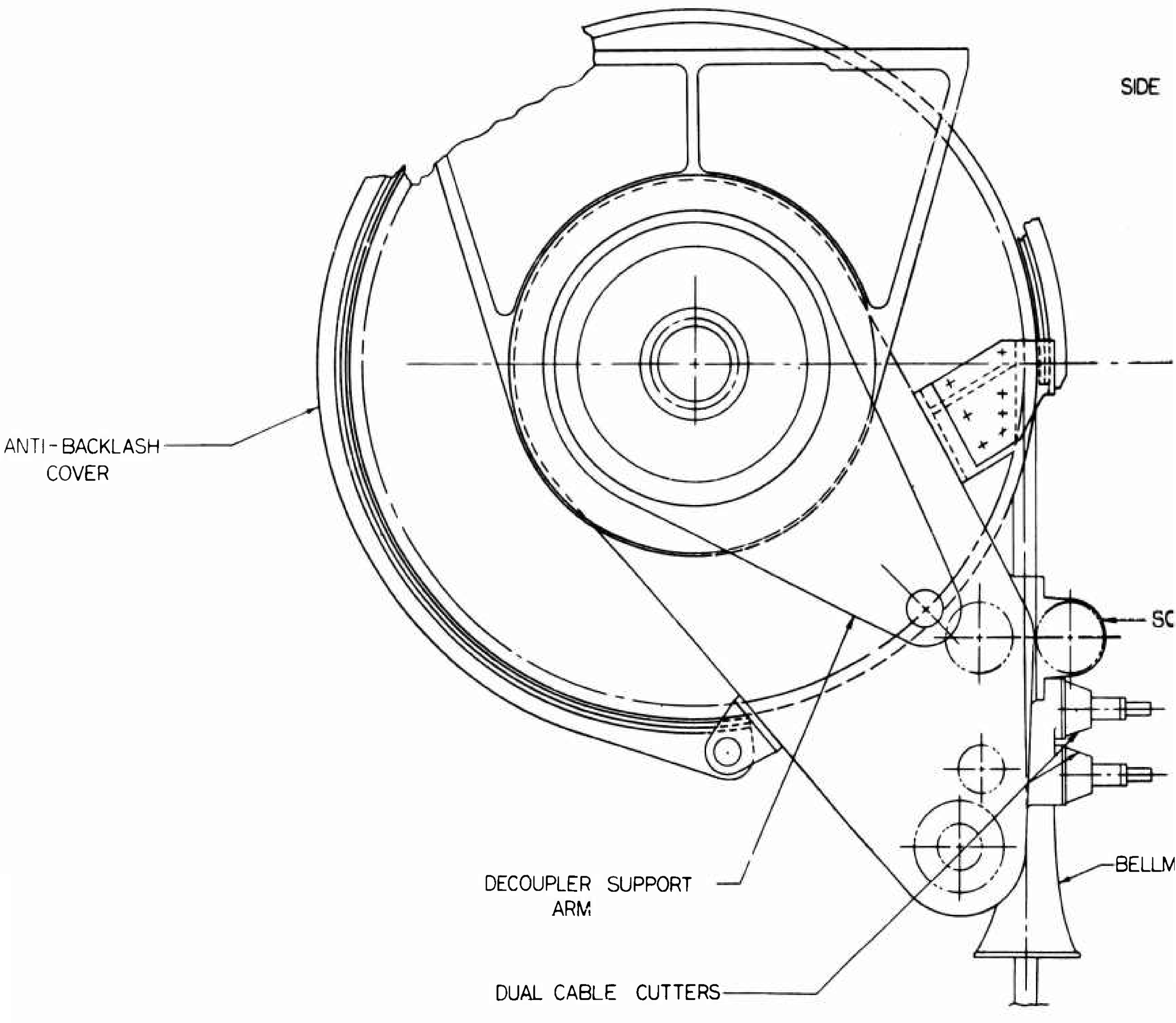
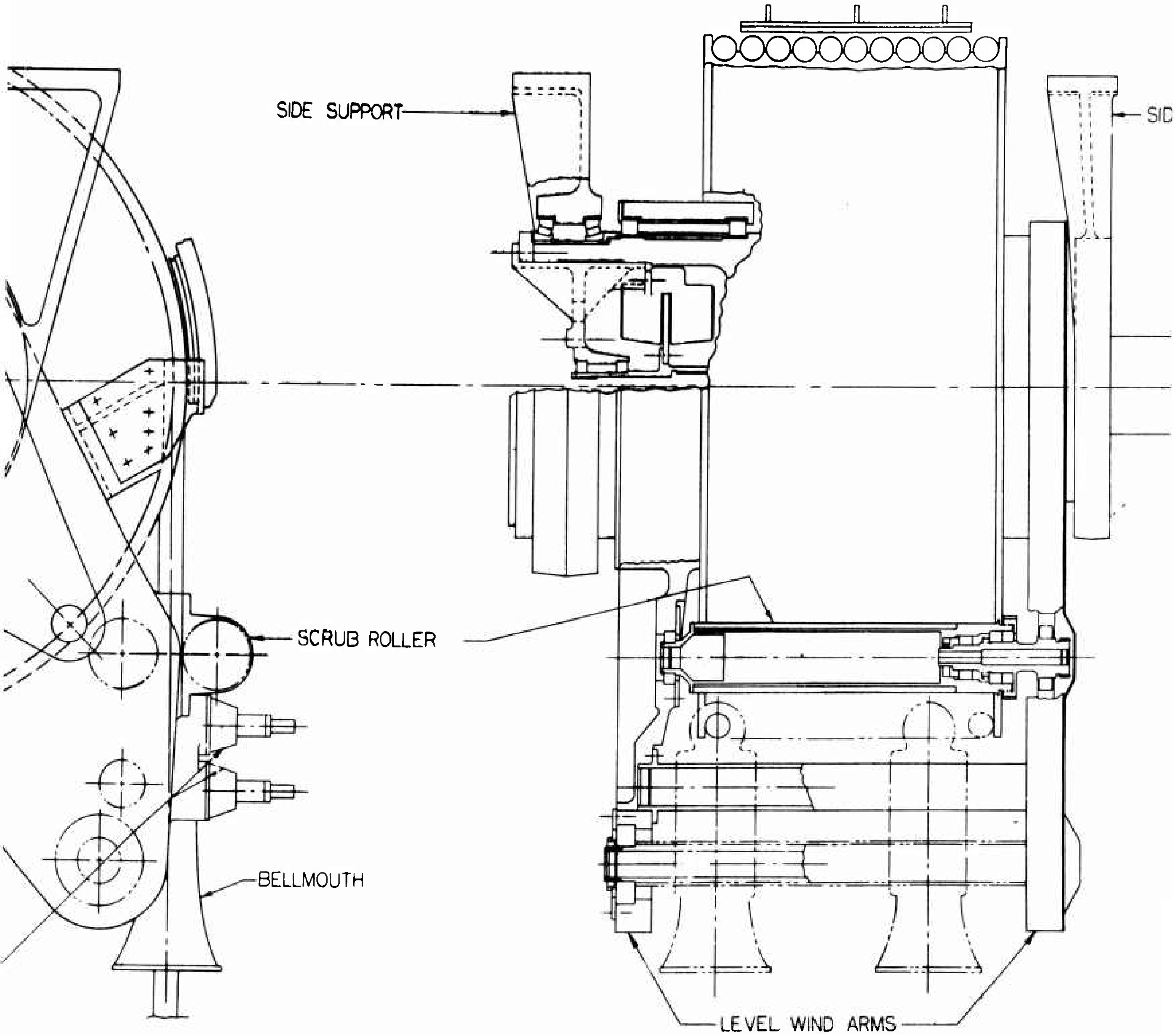
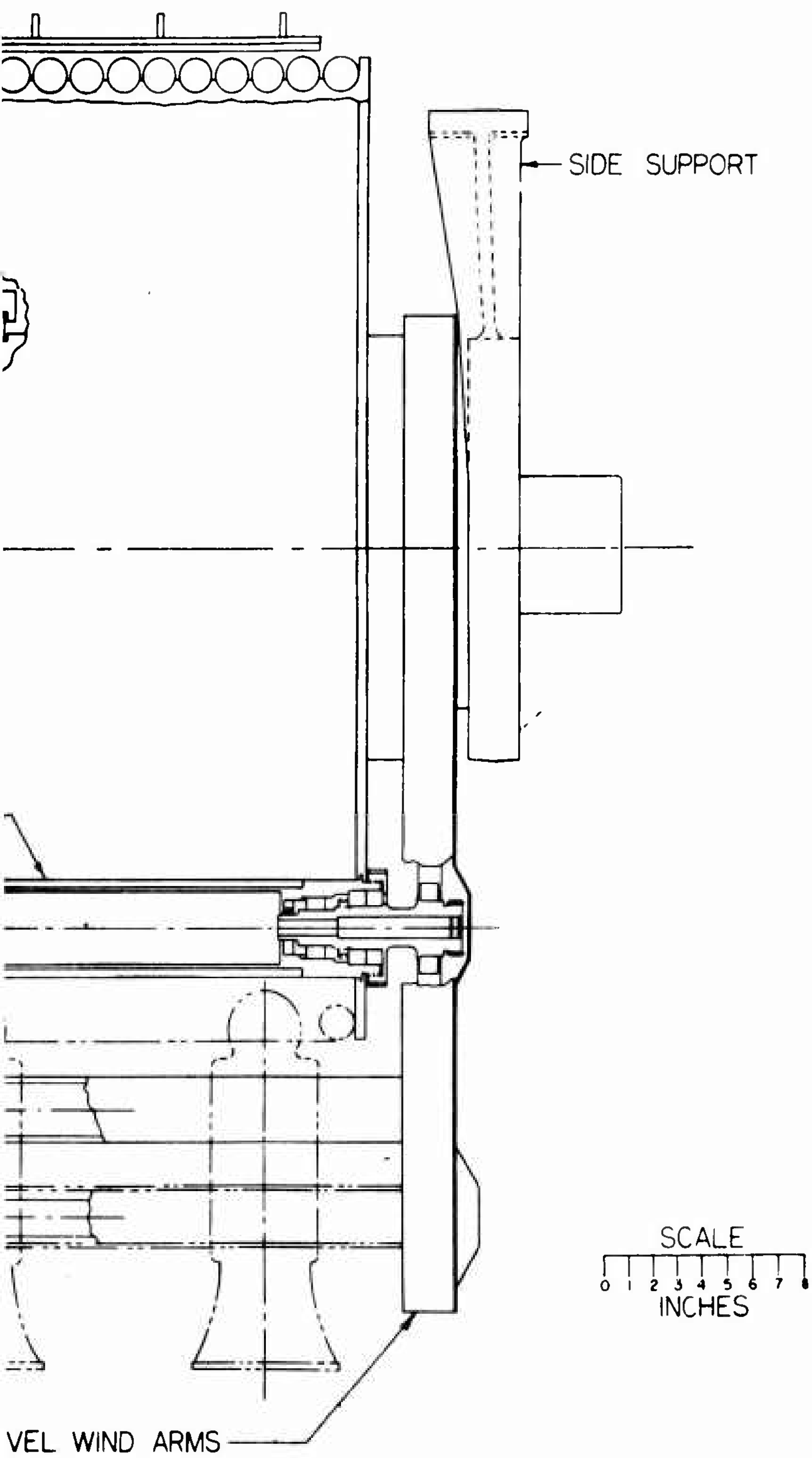


Figure 4. Conventional Design - Single-Point Hoist.

A.



B.



C.

Hoist Location - Tandem-Rotor Aircraft

In the tandem-rotor aircraft (Figure 2, page 7) the single-point hoist is located approximately midway between the rotors to minimize the effect of load oscillations on aircraft stability. It is installed in a well in the fuselage and does not extend below the fuselage when the hook is in the full-up position. This allows pods to be carried by the multi-point hoists without removal of the single-point hoist.

The two-point hoists are located on B.L. 0 at F.S. 413 and F.S. 701. The horizontal spacing of 288 inches was based on the survey of military vehicles in the 15,000- to 40,000-pound weight class. It allows most of the vehicles in this category to be lifted off the ground in the ground pickup mode.

The hoists are located in wells in the fuselage so that they do not extend below the airframe when the hook is in the full-up position. This permits pods to be lifted up and locked to the fuselage. Since the hoists are located relatively high in the fuselage, the effect of lateral load oscillations on aircraft controllability is quite small. Hoist well size requires the addition of approximately 40 pounds to the airframe structure. In addition, this could conflict with the most desirable fuel cell location. No attempt has been made to assess the importance of this conflict in this study.

The four-point hoists are located on B.L. 70 at F.S. 413 and F.S. 701, since this ensures compatibility during ground pickup with most of the vehicles in the 15,000- to 40,000-pound weight category. The hoists are universally mounted on a davit type structure with suitable aerodynamic fairings. Full-up position of the hooks permits pods to be pulled up and locked to the fuselage. As in the two-point system, this relatively high location reduces the effect of load oscillations on aircraft controllability. The universal mounting permits the hoists to be pivoted to reach attachment points on outsized vehicles without inducing heavy side loads on the hoists.

Single-Point Hoist

The conventional (one part, single reeved type) hoist offers the most advantages for application in the single-point location. Because it need not be mounted on a universal-type joint, it can be driven mechanically. It can also fit into a well which has limited vertical space. The drum axis is mounted at right angles to the longitudinal axis of the aircraft, and the level wind assembly is allowed to pivot about the drum axis. This permits large cable angles during tow operations without inducing high loads on the bellmouth. The simplest and most reliable version is the one that requires only one layer of cable wrapped on the drum. Multiple layering is possible but it requires the use of a more complicated, less efficient, and less reliable type of feed screw to ensure even winding of the cable. Figure 4, page 15, shows the conventional, single-layer hoist similar to that used in the CH-54A.

TABLE VI

HOIST LOCATION* - TANDEM-ROTOR AIRCRAFT

	Single-Point Hoist	Two-Point Hoist	Four-Point Hoist
Fuselage Station	557**	413 & 701	413 & 701
Buttock Line	0	0	0
Waterline:			
Longitudinal Cable Swing	165	200	200
Lateral Cable Swing	190	200	200

*Locations described refer to actual cable reaction point

**With one-half usable cable length extended

For the single-rotor aircraft of this study, a maximum cable length of 100 feet can be carried on a single-layer hoist without exceeding desirable lateral cyclic stick movement. The limitation on cyclic stick movement is based on that presently attained in the CH-54A.

For the tandem-rotor aircraft of this study, the single-point hoist drum can be mounted with its axis located parallel to the aircraft longitudinal centerline. With this arrangement, the only limitations in cable length are those imposed by permissible C.G. range and/or hoist well size. This installation, however, restricts the towing capability of the tandem-rotor aircraft from the single-point hoist.

Another type of hoist investigated in this study was the two-part, double-reeved type shown in Figure 5, page 19. This type is used extensively in commercial practice. The two-part, double-reeved hoist has two primary disadvantages when compared to the conventional (one-part, single-reeved) hoist. A cable backlash suppressor will be required to keep the cable from jumping off the drums, both pulleys on the traveling block, and the upper pulley when a load is air dropped. While a suppressor has already been developed for the conventional hoist, the suppressor required for the double-reeved type will be considerably more complex. Cable cutters for the double-reeved hoist have to be mounted in four places to ensure that a sheared cable will not jam up in one of the three pulleys. This requirement for four cable cutters reduces the inherent safety features of the single-point suspension.

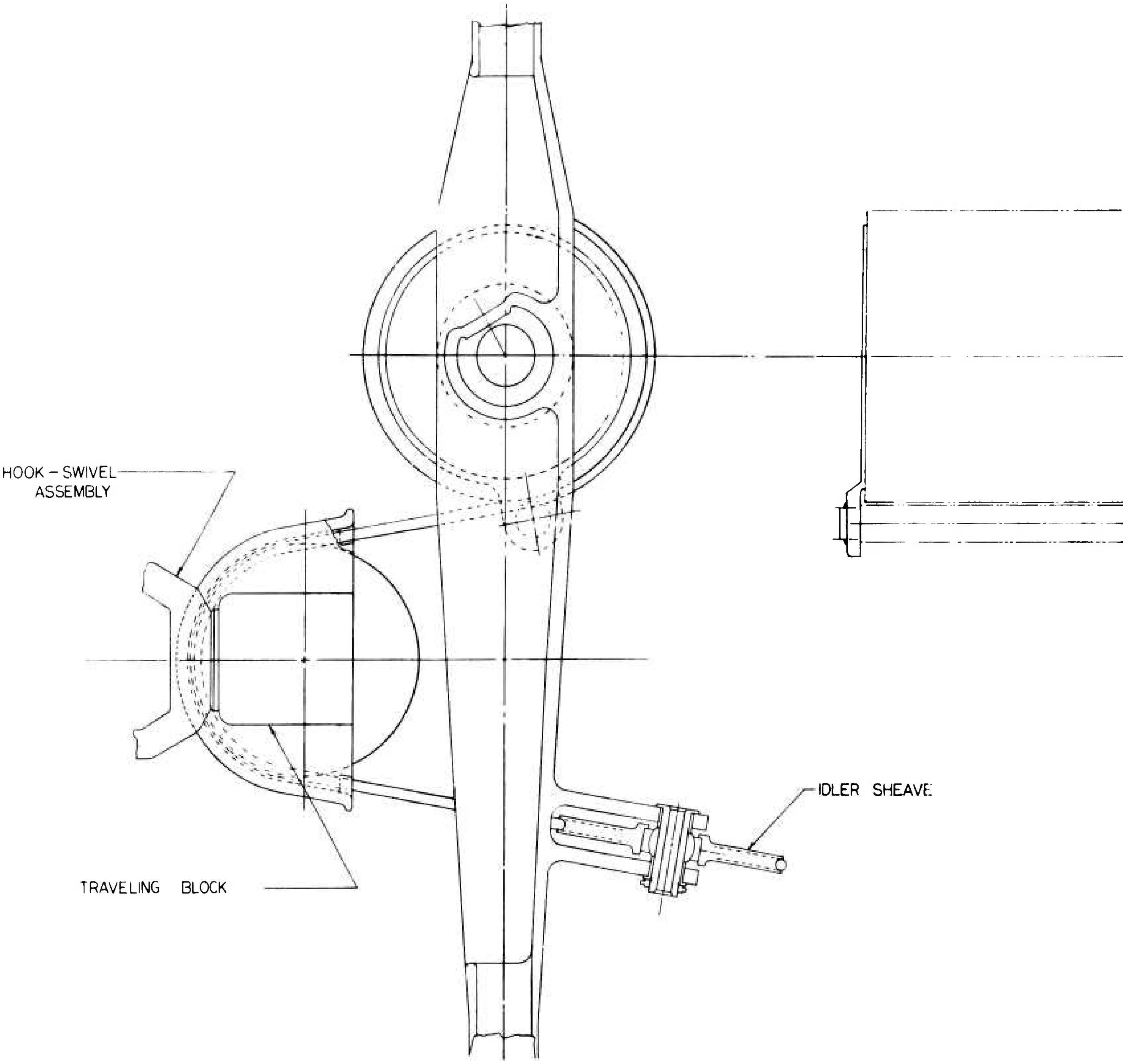
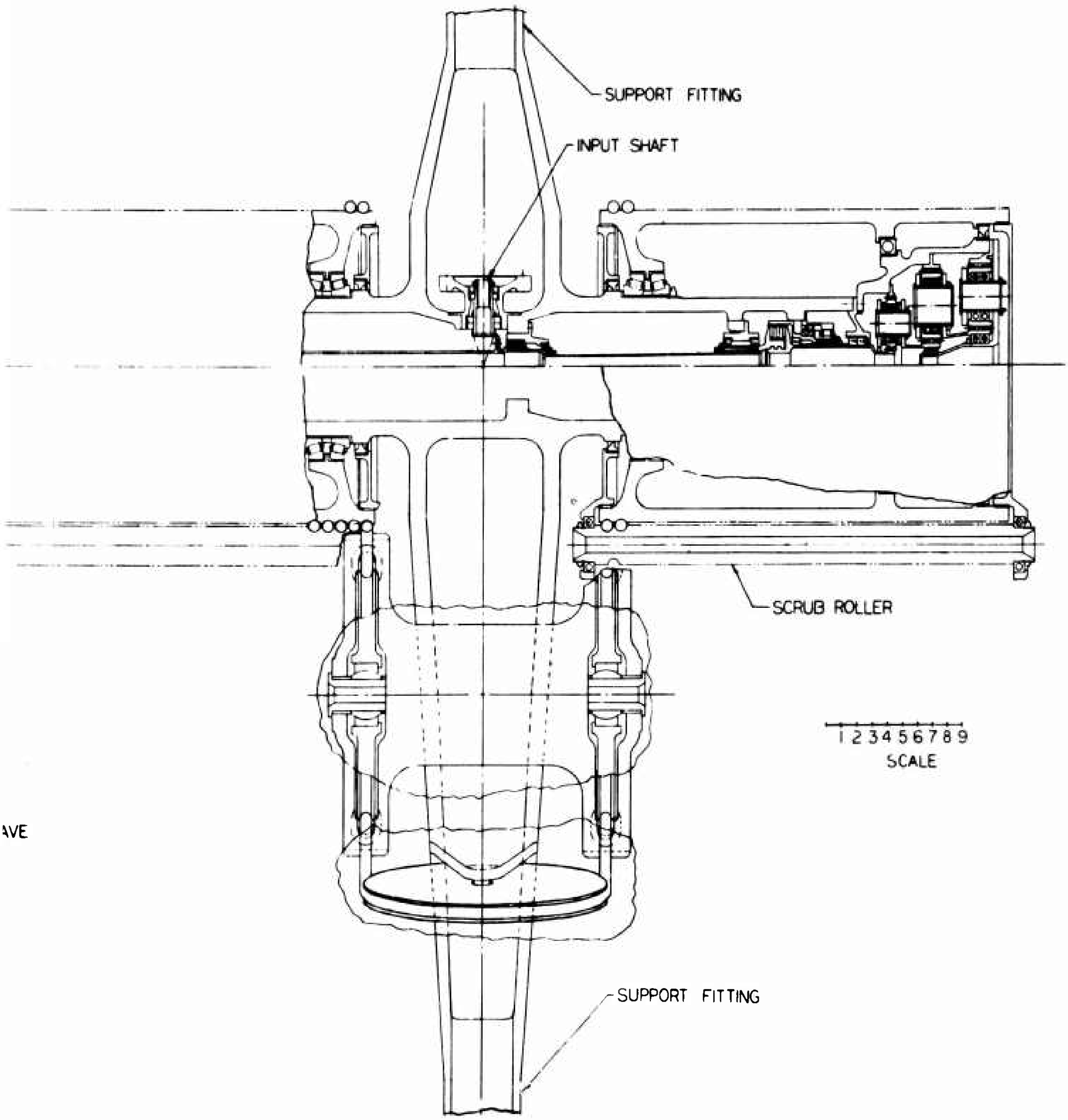


Figure 5. Two-Part, Double-Reeved Hoist.

A.



SUPPORT FITTING
INPUT SHAFT
SCRUB ROLLER
1 2 3 4 5 6 7 8 9
SCALE
SUPPORT FITTING

AVE

B.

The use of a conductor reel to provide a means of transmitting electrical power to the hook adds approximately 50 pounds to the weight of the single-point hoist system. In addition, the use of the reel also reduces the reliability of the system because of the added complexity and the relatively unprotected location of the conductor cable.

Two-Point Hoists

Both the conventional and the zero-moment hoists are applicable to the two-point requirement. A zero-moment hoist is one which maintains the same line of action for the load for all lengths of cable extended and hence always has the same reaction point on the supporting structure.

Since there is no requirement to have the two-point hoists pivot laterally in order to pick up outsized loads, the use of a zero-moment hoist is not mandatory. Its greater vertical dimension, required to mount a load isolator most efficiently, requires the use of a deeper well in the fuselage.

The conventional hoist, as described in the Single-Point Hoist section, fits in a shallower well in the fuselage and is equally adaptable to both mechanical and hydraulic power sources. Therefore, since there is only a limited amount of vertical space available in the fuselage of either the single- or tandem-rotor aircraft, a conventional hoist is the most advantageous.

Due to variations in the size of loads to be carried in the multi-point mode, it is necessary to increase the design load rating of the two-point hoists to account for cable angle. A cable angle (with the vertical) of 30° has been selected to permit variations in longitudinal dimension of load. The required hoist rating is then

$$\text{Rating} = \frac{20,000}{\cos 30^\circ} = 23,100 \text{ pounds} \tag{1}$$

Four-Point Hoists

The use of the zero-moment hoist is mandatory in the four-point system. Its ability to be pivoted to any position allows attachment to a wide variety of cargo sizes and shapes. A mechanical drive system for the zero-moment hoist will be extremely complicated, while hydraulic power offers automatic load equalizing, and, by use of a feedback system (see Figure 13, page 48), permits synchronised operation.

Sikorsky Aircraft experience has shown that a capstan type hoist offers weight advantages only when the long cable lengths are required, as in rescue winch applications. For hoists requiring only 50 feet of cable at low speed and relatively high load capability (such as the H.L.H. multi-point application), the capstan principle offers no advantages; in fact, there may be some penalties in weight and cable life.

A single drum design, universally mounted, offers lighter weight, greater cable life, and somewhat more reliability. In configurations that eliminate the single-point hoist, it is necessary to use three layers of cable in order to retain the zero-moment capability. As discussed in the Single-Point Hoist section, multiple layering requires the use of a more complicated and somewhat less reliable type of feed screw than is needed in a single-layer design. Figure 6, page 23, shows a typical single-layer, zero-moment hoist.

It should also be noted that it is necessary to increase the design load rating above the theoretically possible rating of 10,000 pounds, since it is not possible to lift the wide variety of loads required with this rating. If cable angles (with the vertical) of 30° are utilized to permit the necessary variation in load length and width, a hoist rating of 10,000/Cos 30° = 11,550 pounds is required. Figure 3, page 13, shows the allowable load attachment point space envelope with the selected hoist locations and the 30° cable angle requirement.

HOIST DESIGN

Discussion

The initial consideration in the design of a hoist is the selection of a cable that meets the load and functional requirements (i.e., nonrotating construction with electrical conductors in the core) for this study. Once the cable diameter and the wire size to be used in the cable have been selected, the drum diameter can be calculated. Using standard commercial practice which requires cable drum diameter to be a minimum of 400 times the individual wire diameter, it is possible to determine minimum drum diameter requirements. Other considerations may dictate larger drum diameters, however. In the case of the single-point hoist, the determining factor is the requirement to carry as much cable as possible without exceeding desirable aircraft control limitations (see Single-Point Hoist section, page 17). After the drum diameter has been established, the drum thickness is determined by analysis. Once this is completed, the gearing system can be established and the load brake can be integrated into the primary gear train. The auxiliary gear train is then designed into the level wind arms to provide power for scrub rollers and level wind screw drives. For a single layer of cable, a simple ball screw and nut can be used, whereas in multiple layer hoists, a more complex double helix screw is required.

The accessory drive gearing can then be designed to provide drives for slip rings, cable length indicator potentiometers, and limit switch actuators. Cable cutters, cable backlash clamps (or covers), guide rollers, and limit switches are next integrated into the final design. The support structure is designed to conform to established mounting structures, and load cells and isolator units are designed into the support structure.

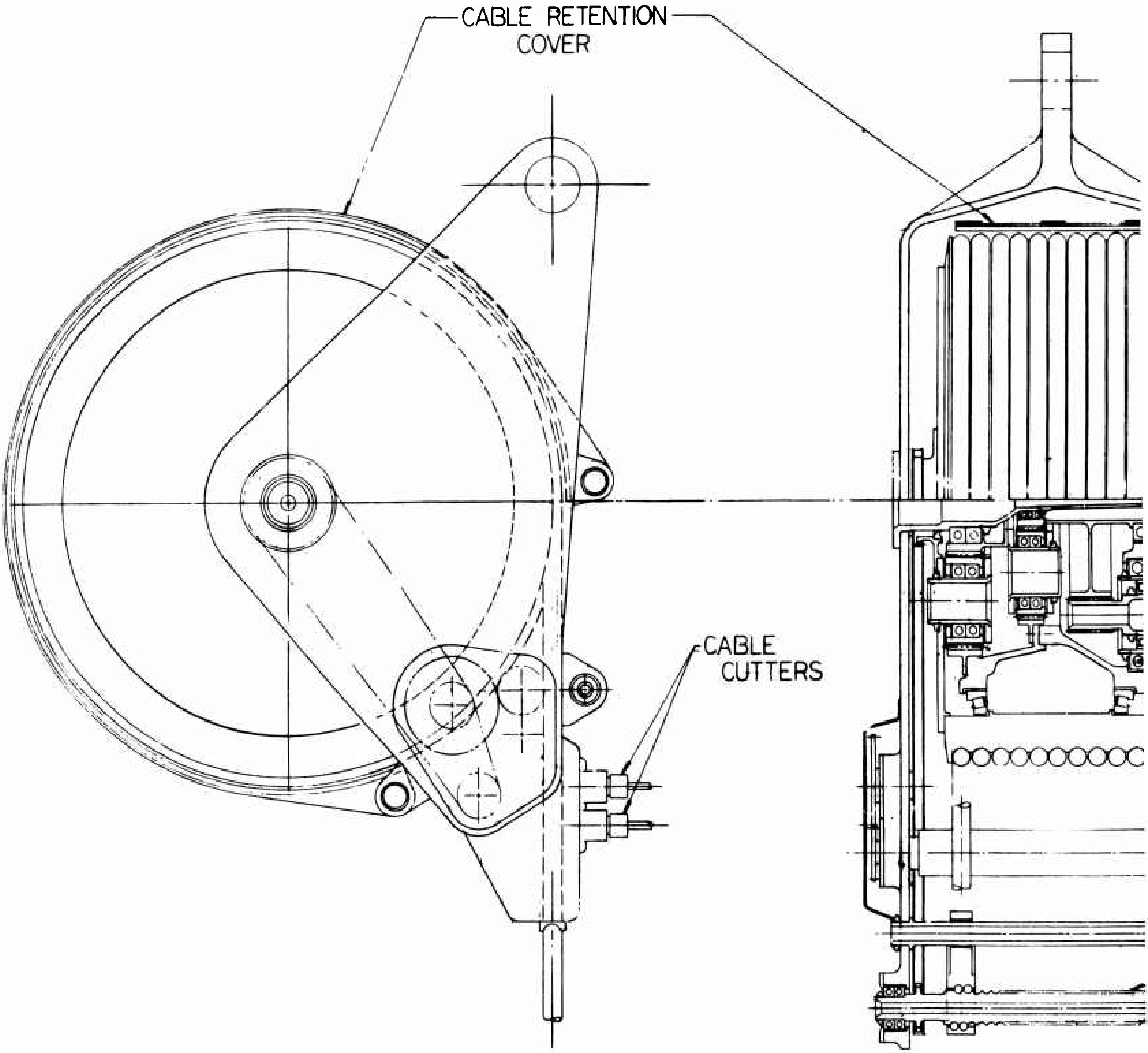
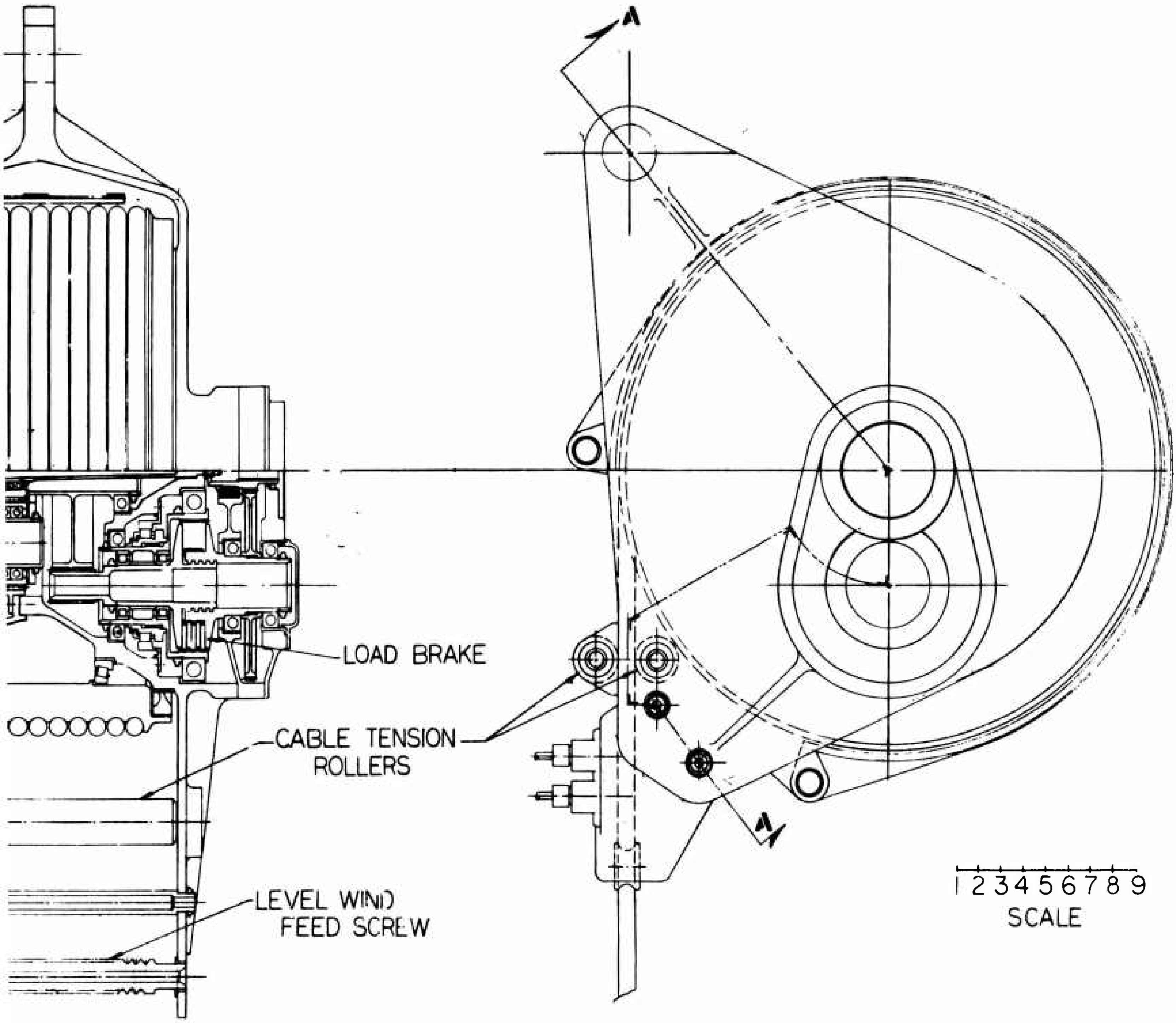


Figure 6. Zero-Moment Hoist.

A.


A
LOAD BRAKE
CABLE TENSION
ROLLERS
LEVEL WIND
FEED SCREW
A
1 2 3 4 5 6 7 8 9
SCALE

B.

Cable Design

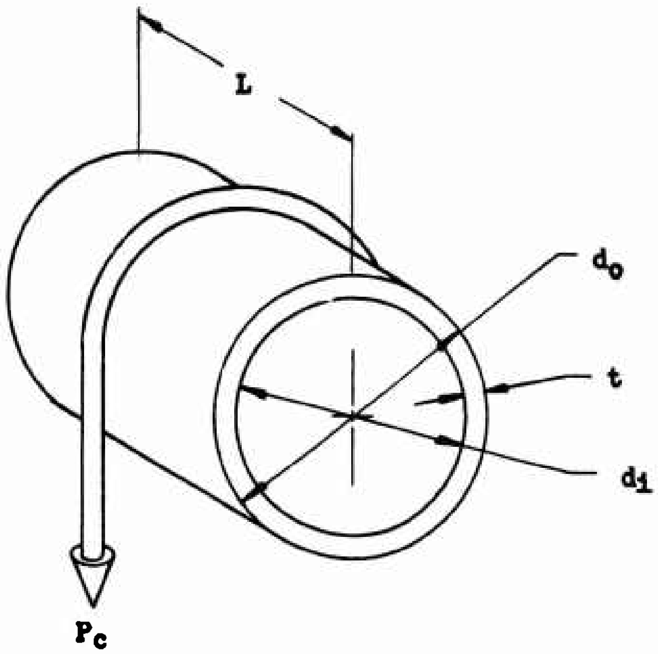
Three cable sizes are needed to meet the requirements for the single-point, two-point, and four-point hoists. All cables are designed to carry seven electrical conductors in a central core and are of nonrotating construction. Five conductors are used to operate the cargo hook; two are spare wires. All are of an extra flexible 18 x 19 construction. A 1.39 diameter is required to meet the 150,000-pound minimum breaking strength requirement of the single-point hoist. A 1.06 diameter cable meets the 97,000-pound requirement for the two-point hoist, and a 0.79 diameter cable meets the 48,500-pound requirement for the four-point hoist.

Since the aircraft's angular accelerations in a maneuver are combined with the linear acceleration effects, the loads at all multi-point hoist attachment points are increased. This load magnification is due to the location of the hoist attachment points some distance from the aircraft's center of gravity. Therefore, an ultimate load factor of 4.2 is required for all multi-point hoists instead of the 3.75 value used for the single-point hoist. Further details on hoist cables are found in the Hoist Cables section, page 49.

Cable Drum Design

The basis for hoist drum analysis for both single and multiple layers of cable is given on the following pages.

Single Layer of Cable:



Stresses in the hoisting drum shell are the result of loads imposed by:

External pressure (P_e) from coiling of ropes under tension

Bending moment (M_b) from rope tension

Torsional moment (M_t) due to power transmission from the gear train to the rope

These loads and stresses can be calculated from the following equations:

$$P_e = \frac{2 \quad P_c}{Pc \quad d_m} \tag{2}$$

$$f_c = \frac{-P_e \ A}{2} \tag{3}$$

where

A is equal to $$\frac{1}{\frac{t}{d_o}\left(1 - \frac{t}{d_o}\right)} \tag{4}$$

$$M_b = \frac{P_c \ L}{4} \tag{5}$$

$$f_b = \frac{M_b}{Z} \tag{6}$$

where

Z is equal to $$\frac{\pi d_o^3}{32}\left[1-\left(1-\frac{2t}{d_o}\right)^4\right] \tag{7}$$

$$M_t = \frac{P_c \ d_m}{2} \tag{8}$$

$$f_t = \frac{M_t}{2Z} \tag{9}$$

For the hoist drums designed in this study, the length-to-diameter ratio is approximately equal to 1. Therefore, for the purposes of finding the approximate magnitude of the compressive, bending, and torsional stresses, the following assumptions can be made:

$$L = d_o = d_m = D$$

Combining equations (2), (3), and (4), we obtain

$$f_c = \frac{P_c}{p_c\, t\, (1 - \frac{t}{D})} \tag{10}$$

Combining equations (5), (6), and (7), we obtain

$$f_b = \frac{P_c\, D}{4\, Z} = \frac{8\, P_c}{\pi D^2 \left[1 - (1 - \frac{2t}{D})^4\right]} \tag{11}$$

Combining equations (7), (8), and (9), we obtain

$$f_t = \frac{P_c\, D}{4\, Z} = \frac{8\, P_c}{\pi D^2 \left[1 - (1 - \frac{2t}{D})^4\right]} \tag{12}$$

By using the previously mentioned assumptions, it can be seen that the bending and torsional stresses are of the same order of magnitude.

In order to compare the magnitudes of the stresses further, the following assumptions can be made:

$$\frac{t}{D} = .038 \qquad D = 25 \qquad t = .95 \qquad p_c = 1$$

These assumptions are within average values used in this study.

Substituting these values into equations (10) and (11), we obtain

$$f_c \approx \frac{P_c}{.914} \tag{13}$$

$$f_b \approx f_t \approx \frac{P_c}{65.76} \tag{14}$$

Therefore, the bending and torsional stresses in the hoisting drum shells used in this study are in the order of magnitude of 70 times less than the compressive stresses and may be neglected when designing the drums.

Material Trade-off Investigation

The following materials have been tentatively selected for use in the drum:

TABLE VII

DRUM MATERIAL MECHANICAL PROPERTIES

Material	F_{tu} Ultimate Tensile Strength	F_{cy} Yield Strength	Density (lb/in.^{3})
7079-T6 Aluminum	72,000	65,000	.099
AZ80A-T5 Magnesium	42,000	25,000	.0652
4340 Steel	180,000	179,000	.283

The drum weight per inch is given by

$$W = t \; d_o \left(\frac{t}{d_o} - 1\right) \rho \pi \tag{15}$$

Since the design is based on compressive stress only, equations (13) and (15) may be solved simultaneously to give

$$W = \frac{\rho \pi d_o^2}{p_c \; d_m} \; \frac{P_c}{f_c} \tag{16}$$

For a given hoist drum, d_o, p_c, and d_m will be constant. Hence, the lightest drum will be the drum with the lowest value of $\rho P_c/f_c$.

The following is a design relationship for cable load:

$$P_c \text{ (ult)} = 1.304 \quad P_c \text{ (yield)} \tag{17}$$

The weight of the drum must be investigated under ultimate and yield conditions. To accomplish this, the following constants can be evaluated:

$$\text{ult} - C_1 = \frac{\rho P_{c \text{ ult}}}{F_{tu}} = \frac{1.304 \rho P_{c \text{ yield}}}{F_{tu}} \tag{18}$$

$$\text{yield} - C_2 = \frac{\rho P_{c \text{ yield}}}{F_{cy}} \tag{19}$$

The values of ρ , F_{tu}, and F_{cy} for different materials may now be substituted into the preceding equations. The lightest drum based on ultimate load conditions will be the drum with the lowest value of C_1. The lightest drum based on yield load conditions will be the drum with the lowest value of C_2. The design will then be based on the load conditions which produce the highest value of C_1 or C_2 for a particular material. Substitution of the values of ρ , F_{tu}, and F_{cy} into equations (18) and (19) for various materials is given below:

1. 7079-T6 Aluminum forging

$$C_1 = \frac{1.304\,\rho\,P_c\ \text{yield}}{F_{tu}} = 1.79 \times 10^{-6}\ P_c\ \text{yield}$$

$$C_2 = \frac{\rho\,P_c\ \text{yield}}{F_{cy}} = 1.52 \times 10^{-6}\ P_c\ \text{yield}$$

2. AZ80A-T5 Magnesium forging

$$C_1 = \frac{1.304\,\rho\,P_c\ \text{yield}}{F_{tu}} = 2.02 \times 10^{-6}\ P_c\ \text{yield}$$

$$C_2 = \frac{\rho\,P_c\ \text{yield}}{F_{cy}} = 2.60 \times 10^{-6}\ P_c\ \text{yield}$$

3. 4340 Steel forging

$$C_1 = \frac{1.304\,\rho\,P_c\ \text{yield}}{F_{tu}} = 2.05 \times 10^{-6}\ P_c\ \text{yield}$$

$$C_2 = \frac{\rho\,P_c\ \text{yield}}{F_{cy}} = 1.58 \times 10^{-6}\ P_c\ \text{yield}$$

Therefore, the 7079-T6 aluminum forging, based on ultimate load conditions, will result in the lightest hoist drum design. The aluminum drum will be 31 pct lighter than a magnesium drum (designed for yield conditions) and 12 pct lighter than a steel drum (designed for ultimate conditions).

Note: The use of a maraging steel of F_{tu} = 250,000 psi will result in a significant weight savings. This design will, however, have a very low thickness to diameter ratio and may present machining problems because of distortion of the thin-walled drum. Maraging steels have also been known to present stress corrosion problems. In addition, the cost is approximately six times that of aluminum. Therefore, high strength steel was not used for the hoisting drum. However, it is worthy of further study as a possible means of weight saving along with the titanium alloys and glass. An evaluation of these materials will be conducted as part of Phase II if time permits.

Drum Thickness Analysis

The equation for compressive stress may now be solved directly for thickness to diameter ratio using F_{tu} for 7079-T6 aluminum:

$$\frac{t}{d_o} = \frac{1}{2} - \sqrt{\frac{36,000 - P_{e\,ult}}{144,000}} \qquad (20)$$

Table VIII summarizes the drum thickness calculations for the single-point, two-point, and four-point hoists with a single layer of cable.

TABLE VIII
DRUM THICKNESS SUMMARY,
SINGLE CABLE LAYER DESIGNS*

		Single-Point	Two-Point	Four-Point
P_c	- Cable load (ult), lb	150,000	97,000	48,500
p_c	- Cable pitch, in.	1.5	1.1875	.875
d_m	- Mean drum dia., in.	38.99	29.69	22.19
P_e	- External pressure (ult), psi	5290	5500	5190
t/d_o	- Calculated	.0382	.0398	.0376
d_o	- Effective drum OD, in.	37.806	28.905	21.568
t	- Drum thickness, in.	1.44	1.15	.81
W	- Weight of drum, lb per in.	16.40	10.03	5.22

*Drum Material: 7079-T6 Aluminum Alloy

Multiple Layers of Cable

Since the external pressure caused by coiling of the cable under constant tension is the major factor in the design of drums considered in this study, it is important to determine this pressure accurately. When there is more than one layer of cable on the drum, the pressure increases, but not as a direct ratio of the number of layers. The outer cables tend to compress the inner cables and drum, thereby relieving some of the pressure originally caused by the inner cables. The general equation for the pressure on the drum is given by

$$P_e = \frac{2\,K\,P_c}{P_c\,d_m} \tag{21}$$

where

K is a factor less than n_1

(number of cable layers)

For $n_1 = 1$, $K = 1$. K can be calculated from

$$K = \sum_{j=1}^{j=n_1} 1 - \frac{2\,A\,E_L}{P_c\left[d_o + d_c \sin\gamma(2j-2)\right]}\left\{\frac{1}{E_d}\left[\frac{d_o^2 + d_i^2}{d_o^2 - d_i^2} - \nu\right] x \right.$$

$$\left.\left[K - \sqrt{\frac{j(K^2-1) + (n_1 - K^2)}{n_1 - 1}}\right] + \frac{2d_c(n_1 - j)}{E_t(n_1-2)}\left[\frac{(n_1-j-1)\,K}{(n_1-1)\,d_o} + \frac{(j-1)}{d_o + d_c \sin\gamma(2j-4)}\right]\right\} \tag{22}$$

where the last term is equal to 0 for $n_1 < 3$ (term with E_t in denominator)

The trial and error solutions to this equation for single and multi-point multiple layer hoist drums are presented in Table IX.

TABLE IX
DRUM THICKNESS SUMMARY,
MULTIPLE CABLE LAYER DESIGN

		Single-Point 2 Layers	Single-Point 3 Layers	Two-Point 3 Layers	Four-Point 3 Layers
P_c	- Cable load (ult), lb	150,000	150,000	97,000	48,500
p_c	- Cable pitch, in.	1.5	1.5	1.1875	.875
d_m	- Drum mean dia, in.	38.99	38.99	29.69	22.19
d_o	- Drum effective OD, in.	37.806	37.806	28.905	21.568
K	- Drum pressure constant	1.495	1.939	1.961	1.930
P_e	- External pressure, psi	7,670	9,950	10,790	9,740
t/d_o	- Calculated	.0565	.0747	.0816	.0730
t	- Drum thickness, in.	2.14	2.82	2.36	1.57
W	- Drum weight, lb per in.	23.978	31.026	19.681	9.887

Hoist Gearing Configurations and Load Brake

Three different gear arrangements were considered. Table X summarizes the overall gear ratios required for the single- , two- , and four-point hoists using a high- and low- speed input drive.

TABLE X
HOIST GEAR RATIOS

	Single-Point	Two-Point	Four-Point
High-speed input, rpm	6000	6500	7200
Low-speed input, rpm	3000	2500	2750
Output rpm	6.1	4.02	5.34
Overall ratio (high)	983.6	1616.9	1348.3
Overall ratio (low)	491.8	621.9	515.0
Min. stages of gearing (high RR)	5	5	5
Min. stages of gearing (low RR)	4	4	4

As can be seen in Table X, four gear stages are required for the low input drive speeds, while five gear stages are required for the high input drive speeds. These numbers refer to conventional gear stages such as a spur gear and pinion or a conventional planetary arrangement (sun gear driving, ring gear fixed, cage driven). If a compound planetary is used, the number of stages can be reduced by one or two because of the higher ratios obtained.

In general, a spur mesh will be lighter for low torque and a planetary will be lighter for high torque applications because of the load splitting capabilities. All of the hoist designs use a Weston brake for controlling the load. This type of brake holds the load when the power source is shut off, locks as a unit when raising the load, and slips at the same speed as the driver when lowering the load. The Weston brake should be located in the gear train a minimum of one gear stage from the input drive. This is to assure locking in the event of a mechanical failure at the drive source, which is usually the weak link. The small drag provided by the first stage will prevent load runaway by providing the torque necessary to lock the brake plates when lowering the load. Since the Weston brake is a purely mechanical device, the load can be maintained in the event of a hydraulic failure or a mechanical failure of the drive train which occurs before the brake input.

The following discussion refers to a hoist utilizing low-speed inputs. For high-speed inputs, another planetary stage can be added. Figure 7 shows a hoist arrangement utilizing three conventional planetary stages and a spur gear input stage.

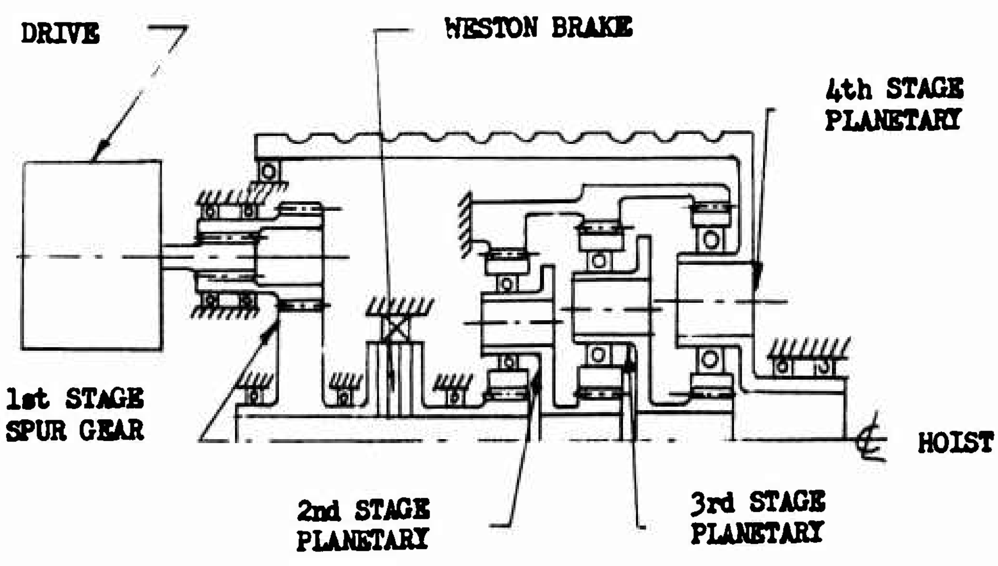


Figure 7. General Gear Arrangement - One Spur and Three Planetary.

In the arrangement shown in Figure 7, it is required that the motor be located off the centerline of the hoist. This may cause interference problems with the drum and motor mount. Figure 8 shows another gear configuration using two spur gear meshes and two conventional planetary stages. In this arrangement, the drive is located on the hoist centerline, and the two side plates can be rigidly connected.

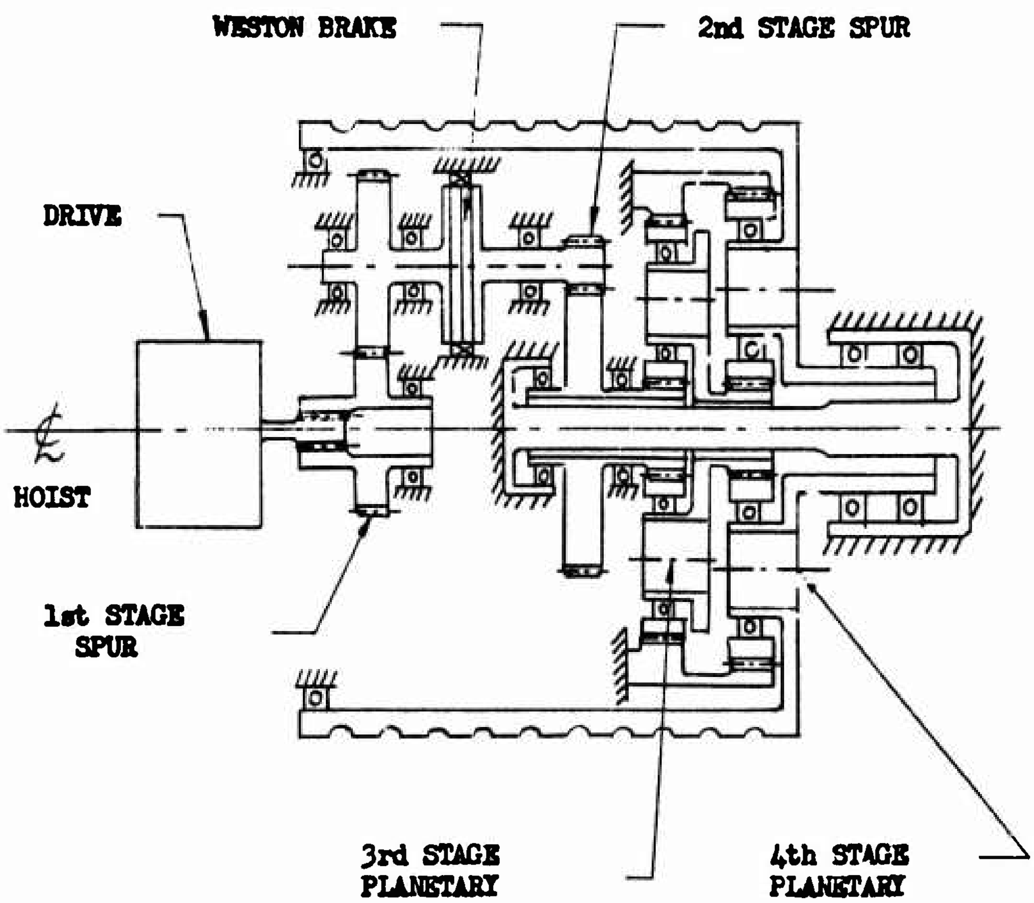


Figure 8. General Gear Arrangement - Two Spur and Two Planetary.

Figure 9 is a compound planetary driven hoist arrangement. It can be designed with reduction ratios of 3:1 to 150:1. A design of this type can replace three conventional planetary stages at a 40 pct weight savings. Its disadvantage is that its efficiency is somewhat lower than that of three conventional planetaries. A 131:1 compound planetary was designed with an efficiency of approximately 92 pct, while the efficiency of an equivalent system using three conventional planetaries was approximately 98 pct.

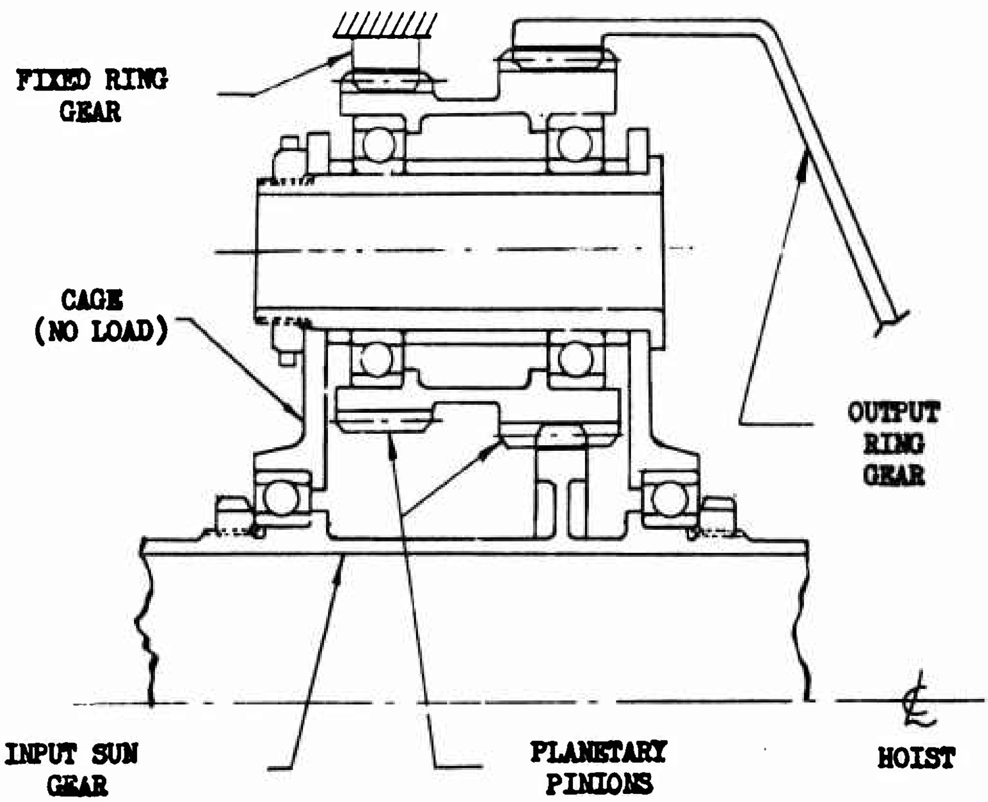


Figure 9. General Gear Arrangement - Compound Planetary.

Figure 10 shows a general gear arrangement for a compound planetary with a conventional planetary for the first stage. This arrangement is the lightest considered, but approximately 6 pct in efficiency is sacrificed.

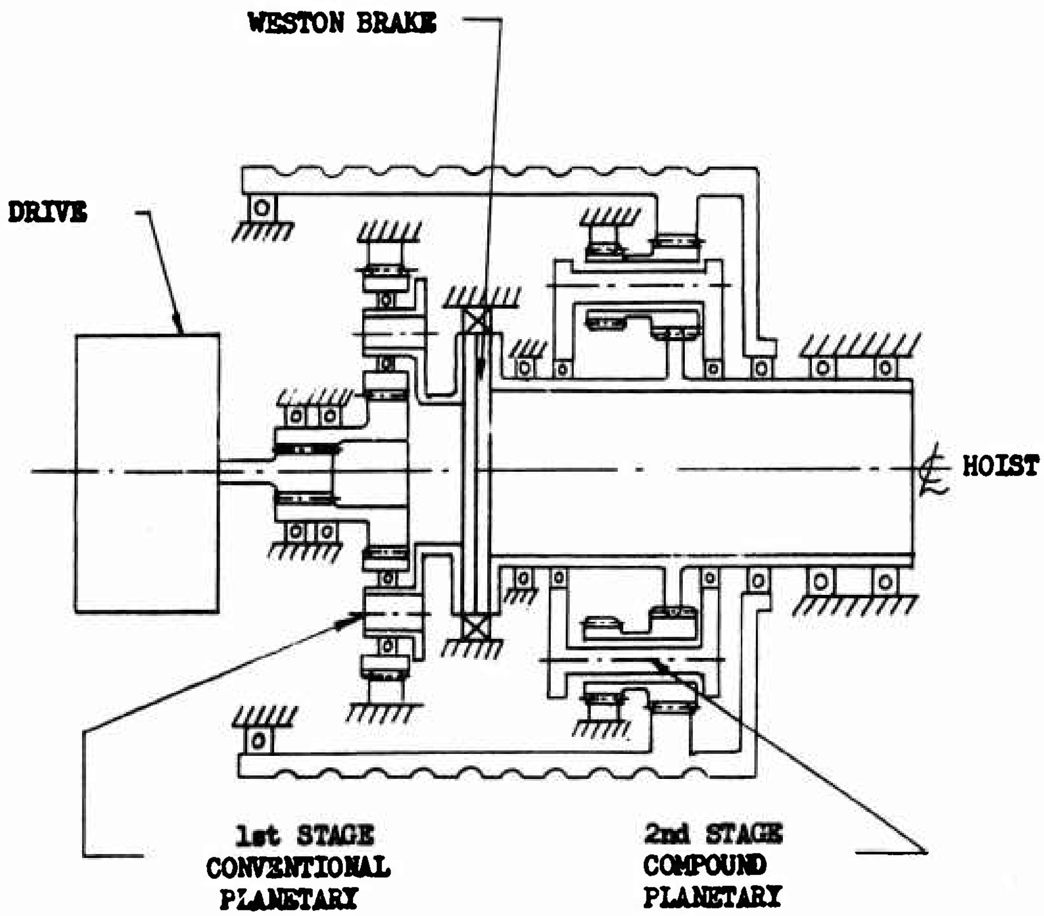


Figure 10. General Gear Arrangement - Conventional and Compound Planetary.

TABLE XI
SUMMARY OF SINGLE-POINT CARGO HOISTS

Type	Designation	Form	Capacity (Pounds)	Cable			Cable Speed (FPM)	Drum Dia. (Inch)	Estimated Weight* (Pounds)
				Length (Feet)	Dia. (Inch)	No. of Layers			
Single-Point Hoist	-	Conven.	40,000	150	1.39	1	60	37.6	2483
	-	Conven.	40,000	150	1.39	3	60	37.6	2260
	A	Conven.	40,000	80	1.39	1	60	37.6	1960
	AA	Conven.	40,000	100	1.39	1	60	37.6	2063
	B	Conven.	40,000	150	1.39	2	60	37.6	2322
	C	Dual Drum	46,200	150	1.06	3	60	28.6	2512
	D	Dual Drum	46,200	80	1.06	2	60	28.6	2138
	E	Two Part Double Reeved	46,200	150	.79	1	60	20.0	2179

*Power source not included in estimated weight

TABLE XII

SUMMARY OF MULTI-POINT HOISTS

Type	Designation	Form	Capacity (Pounds)	Cable Length (Feet)	Cable Dia. (Inch)	Cable No. of Layers	Cable Speed (FPM)	Drum Dia. (Inch)	Estimated Weight* (Pounds)
Two-Point Hoists	F	Conven.	23,100	50	1.06	1	30	28.6	1022
	G	Zero Mom.	23,100	50	1.06	1	30	28.6	1002
	H	Zero Mom.	23,100	88	1.06	2	60	28.6	1072
	J	Conven.	23,100	165	1.06	3	60	28.6	1322
Four-Point Hoists	K	Zero Mom.	11,550	50	.79	1	30	21.4	500
	L	Zero Mom.	11,550	88	.79	2	60	21.4	550
	M	Zero Mom.	11,550	165	.79	3	60	21.4	680
	N	Dual Drum	11,550	2 @ 56	.79	3	60	21.4	990
	P	Conven.	11,550	50	.79	1	30	21.4	488

*Power source not included in estimated weight

Summary

A complete summary of all the cargo hoist types evaluated in this study is given in Tables XI and XII, pages 37 and 38. The weight given includes the weight of hooks and cables but omits the weight of the power source, since both mechanical and hydraulic power sources were considered.

The single-point hoist designated "E" was included since it offers a modified zero-moment capability and a significant weight savings. Its disadvantages, as discussed on page 18, outweigh these advantages; hence, further study of this type is not considered necessary. All hoists have been designed to permit removal from the aircraft without removal of the power source. Gear drives, hydraulic motors, and lines will remain with the aircraft.

POWER SOURCES

Hydraulic

Hydraulic motor drive for either the two- or four-point hoists is possible with existing hardware. However, the pump and motor required for the single-point hoist will require some development. This development and modification is required to adapt to normal temperature environmental operation, since these units are presently designed for the extreme temperature environments on the XB-70. The motor will be a modification of a pump used in this aircraft. The development effort could be reduced if two smaller motors, geared to a common input shaft, were used in place of a large single motor drive. All hydraulic motor combinations offer variable speed control for the single-point hoist. The pump required for configurations that use hydraulic power for the multi-point hoists only, while not a production unit, is of conventional design and hence should not require any development effort.

TABLE XIII
HYDRAULIC PUMP SUMMARY

Application	HP Output	RPM	Disp. (cu in./rev)	Flow (gpm)	Weight (lb)
Single and Multi-point	150	3500	5.873	68	55
Multi-point only	100	4000	2.80	45	35

Both high- and low-speed hoist motors were considered in the preliminary system evaluation in order to determine which type would best meet the requirements. Preliminary calculations indicated that the weight advantage offered by the high-speed motors was offset by the weight of the added transmission system drive gearing. Since weights are nearly equivalent, the low-speed motors were selected because they offered a greater reliability and longer life as compared to the high-speed designs. The motor selected for both traction sheaves and the conductor reel is a standard, off-the-shelf component.

TABLE XIV
HYDRAULIC MOTOR SUMMARY

Hoist Designation	HP Output	RPM	Disp. (cu in./rev)	Flow (gpm)	Weight (lb)
A, AA, B, C, D,	104	3000	5.25	68	44
E	104	6000	2.70	68	24
H, J, N	70	4050	2.50	45	22
A, B, C, D, E	52*	4400	1.80	34	19
F, G,	35	2500	2.35	22.5	22
L, M,	35	6500	0.95	22.5	11
K, P,	17.5	2750	0.95	11.1	10
	17.5	7200	0.367	11.4	5
Conductor Reel Traction Sheaves	1.0	1700	0.095	0.70	2.6

*Two motors required per hoist

Mechanical

All mechanical drive systems for the single-rotor helicopter are driven by an auxiliary power plant and rotor powered accessory gearbox. This permits ground operation without the rotors turning. In flight, the accessory gearbox is shaft driven by the main gearbox. An identical system is used to power auxiliary drives on the CH-53A. Another version, in which the auxiliary power plant drives the accessory gear train in the main gearbox during operation, is used on the CH-54A. These concepts facilitate ground check-outs of all systems, since a pilot is not required to "run up" the aircraft. In a tandem-rotor aircraft, similar design principles can be utilized to permit this type of operation.

A clutch-reverser unit mounted on the gearbox is used to provide the opposite shaft rotation required for raising and lowering. By actuating both clutches, the mechanical drive system can be disengaged from the accessory

gearbox when hoist operation is not required. An alternate type of power takeoff unit is described in Appendix II. It was not used in this phase of the study because it is not a fully developed unit. The concepts, however, are now being used in similar units for constant speed drives in several operational aircraft. The mechanical variable speed drive offers variable speed drive for the mechanical system and is considered worthy of further study.

The angle gearboxes and drive shafts utilized follow standard Sikorsky Aircraft design practice similar to that utilized in tail rotor drive system gearboxes and tail drive shafts. No development problems are anticipated for these units.

The individual hoist clutches required in the mechanical drive versions for the multi-point hoists follow standard automotive and marine practice. They are multiple disc types in which the actuation force is supplied by oil at 250 psi. This oil, provided by an accessory gearbox mounted pump, is also used to cool the clutch plates.

Auxiliary Power Plant (APP)

All the external cargo handling systems considered in this study will be powered either hydraulically or mechanically from the aircraft accessory drive gearbox. This unit is driven from the primary rotor drive train when the rotor system is operating and from an auxiliary power plant (APP) on the ground to permit ground check-out and acquisition of loads when the rotor system is locked.

A separate gas turbine as the sole source of power for the hoist systems was considered and was rejected because it was heavier and less reliable. For an APP driven cargo system, an APP with a hot day output power ranging from 100 to 170 horsepower (reference Tables XVII and XVIII, pages 87 and 89) depending on the system configuration, is required. In addition, this system requires either continuous operation of the APP in flight or restart for raising or lowering cargo at the acquisition or release site.

Electrical

There are no electric motors of aircraft quality available in the 100-horsepower class required for the single-point hoists nor in the 35-horsepower class required for the two-point hoists. A motor is available in the 17.5-horsepower class required for the four-point hoists. However, its weight of 17.5 pounds, compared to 5 pounds for a similar high-speed hydraulic motor, plus the requirement for large electrical lines and considerably larger alternators, would require a considerable weight increase. Therefore, an electrical motor drive was not considered feasible for the external cargo handling system power source.

CLUTCH-REVERSER UNIT

A clutch-reverser unit is mounted on the aircraft accessory drive gearbox and transmits power in either direction of rotation to the hoists. It consists of twin reversing clutches. By declutching both of the reversing clutches, the mechanical drive system can be disengaged from the power source when hoist operation is not required. Oil required to actuate the piston which clamps up the plates in these clutches is supplied at 250 psi by a pump mounted on the accessory gearbox. By controlling the pressure rise to 250 psi, smooth, shock-free operation is obtained. This same oil supply (transmission oil) is also used as a coolant for the clutches when they are disengaged. The use of a clutching arrangement, which can be uncoupled from the power source, is possible because each hoist incorporates an automatic load holding brake. By providing smooth, shock-free acceleration of loads, the need for a variable speed drive can be eliminated.

In the event that it becomes mandatory to provide variable speed operation of the single-point hoist, the clutch-reverser unit can be replaced by a toroid drive unit of the type described in Appendix II. While units of the size required for this application have not been built, the design concept has been proven by use of smaller units for constant speed drives presently installed in the Navy A4E.

CONTROL SYSTEMS

Discussion

The control system employed for both single and multi-point hoists is dependent upon the type of power source utilized for each system. For each of the two power sources considered in this study (mechanical and hydraulic), a separate control system approach must be devised. Basically, the mechanical drive concepts require mechanical clutching operations, while the hydraulic drives are flow controlled. In the multi-point systems, the mechanical drive concepts require electromechanical feedback, while the hydraulic drives employ both electromechanical and hydromechanical feedback to provide load equalization and/or synchronization. All systems utilize electrical control of mechanical and/or hydraulic components in order to enable control functions to be accomplished by the pilot, the copilot, and a dismounted crew member.

Single-Point Hoist Control System - Hydraulic Power Source

A single pump mounted on the accessory gearbox supplies the hoist motor. This pump is also used to supply the four-point hoists in the -2 configuration. The hydraulic system is a pressure demand type in which the rate of flow is varied by controls external to the pump. This control is achieved by a torque motor controlled servo which varies pump displacement. The pressure developed is only that required by system loading plus line losses. This system was chosen to eliminate heat generation inherent in systems using a pressure compensated pump at anything less than full load.

Single-Point Hoist Control System - Mechanical Power Source

A power takeoff shaft is used to drive the hoist. A clutch-reverser unit mounted on the accessory gearbox enables the hoist to raise or lower the load. It also permits disengagement of the power source at which time the load brake in the hoist maintains the cable position.

The clutch-reverser unit is similar to the types widely used in marine applications in small power boats. It utilizes oil-actuated, multiple-disc clutches. The oil is supplied at 250 psi by an accessory gearbox mounted pump. Appropriate solenoid operated valves direct the flow of oil to the proper clutch.

Two-Point Hoist Control System - Hydraulic Power Source

A single pump mounted on the accessory gearbox supplies the hoist motors. The hydraulic system is a pressure demand type in which the rate of flow is established by controls external to the pump and the pressure is determined by the line losses and the load. It is a closed system with a pump supercharge of 50-100 psi. Fluid from the pump is delivered to the appropriate subsystem and is then returned directly to the pump inlet. A separate replenishment pump system supplies fluid to replace that lost from the closed hoist system due to pump and motor leakage and bypass cooling flow. It also provides pressure for pump control.

A feedback control system is used to provide synchronized lifting. This system is described schematically in Figure 11, page 44. The division of flow between the forward and aft hoists is established by servo controlled flow dividers. The signal to the flow dividers is derived from a comparison of the signals from rotary potentiometers mounted on each hoist. Operation of a single hoist is accomplished by supplying a bias signal to the flow dividers to block flow to one of the hoists. Clutches are provided to disengage the potentiometers from the hoists during beeping and are reengaged for collective operation. This enables the cables to be adjusted for any extremes in cargo shape and then permits the established cable lengths to be maintained during collective operation. This system eliminates errors due to differences in motor efficiency and initial settings of the divider valves. The maximum error is estimated to be in the order of 3-7/8 inches in 50 feet.

It will be possible to deenergize the feedback system during the initial stages of hoisting, thus permitting the basic system, which is inherently load equalizing, to equalize loads on the cables automatically. This method of hoisting is feasible only if the lifting points are located symmetrically about the center of gravity (C.G.) of the load. If the C.G. is not symmetrically located, the vehicle could assume an unsafe attitude. Consider, for example, a vehicle with its C.G. located at a point 40 pct of distance between the pickup points to be lifted with 30 feet of cable extended. The vehicle would then assume a nose-down attitude of 43° when the cable lengths were adjusted to maintain equal cable loads. Figure 12, page 46, shows resultant load attitude, with varying C.G. locations, resulting from the use of an automatic cable load equalizing feature. There-

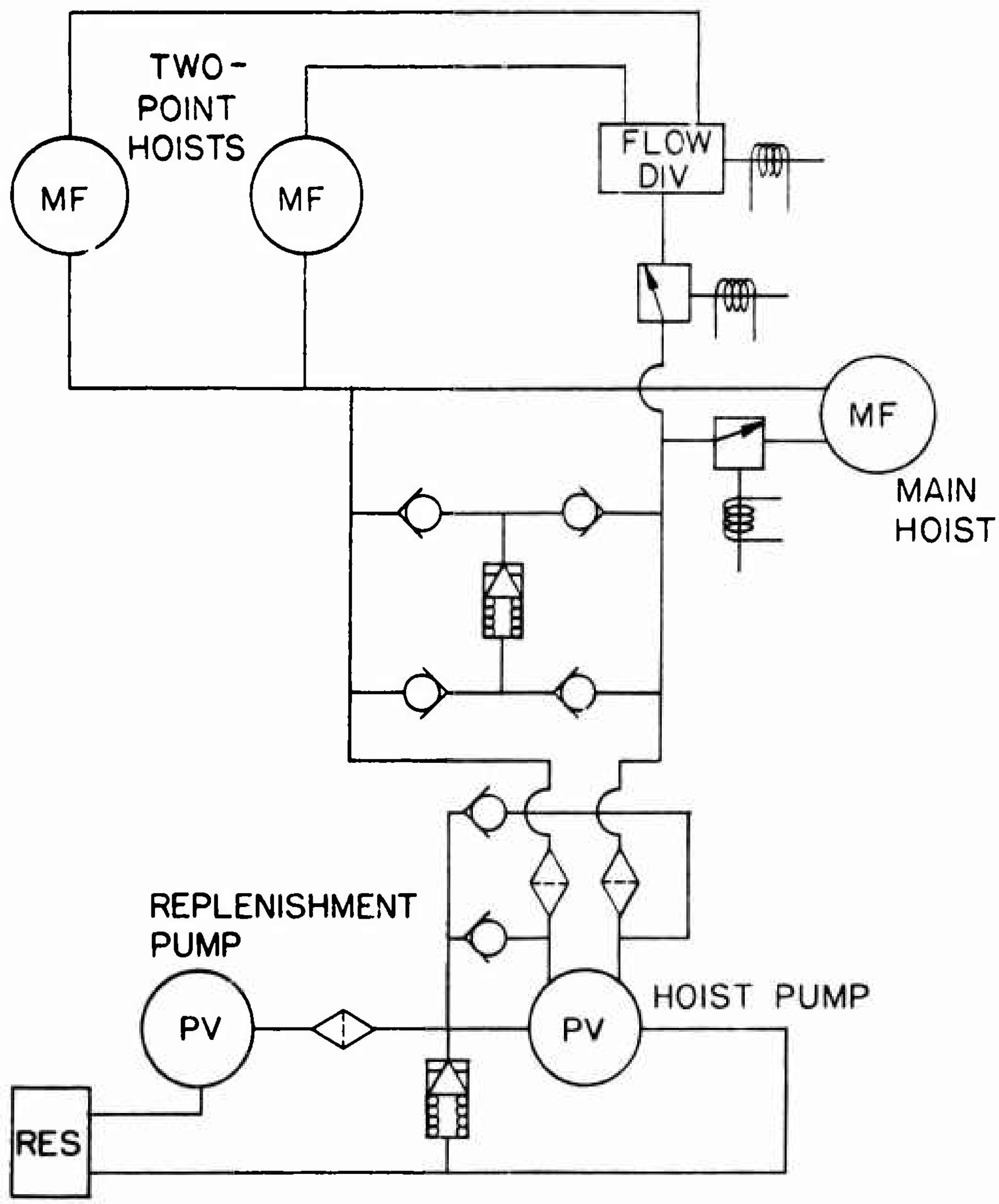


Figure 11. Hydraulic Schematic - Single-Plus Two-Point Suspension.

fore, in the lifting operation of the above example it is desirable to keep the feedback system energized to maintain a level attitude of the load.

Two-Point Hoist Control System - Mechanical Power Source

Power takeoff shafts are used to drive the hoists. A clutch-reverser unit mounted on the accessory gearbox enables the hoist to raise or lower the load. A separate clutch is used to disengage the single-point hoist, and clutches are used to disengage either the forward or aft hoists as required.

Synchronized operation is attained by engaging both forward and aft clutches and then engaging the clutch-reverser unit. This operation is automatically sequenced so that only one control motion will be required.

Equal cable loading is attained by disengaging the clutch driving the heavily loaded hoist and allowing its load brake to maintain the load while operating the lightly loaded hoist until equal cable loads are attained. Load indicators will make it possible to determine when equal loading is obtained. A more elaborate feedback system, utilizing the output of the load cells, could be devised to accomplish automatic load equalizing. Development of such a system is feasible but probably not warranted, since, by operating the hoists individually and using the load indicators to equalize cable loads, the same result can be obtained with considerably less complication.

Four-Point Hoist Control System - Hydraulic Power Source

In one of the separate function systems investigated, incorporating single- and four-point hoists, a single pump mounted on an accessory gearbox supplies the hydraulic power to both. The systems will be isolated from each other and from the pump by electrically operated shutoff valves. For the hydraulically powered combined function systems, a 68 gpm, 3500 psi pump is the source of power. Two of the separate function systems that utilize mechanical drives to the single-point hoist require a 45 gpm, 3500 psi pump as a power source for the hydraulically driven multi-point system.

The hydraulic system is a pressure demand type in which the rate of flow is established by controls external to the pump and the pressure is determined by the line losses and the load. It is a closed system with a pump supercharge of 50-100 psi. Fluid from the pump is delivered to the appropriate subsystem and then returned directly to the pump inlet. A separate replenishment pump supplies fluid to replace that lost from the closed system due to pump and motor leakage and bypass cooling flow and also provides pressure for pump control. A feedback control system is used to provide synchronized lifting. This system is described schematically in Figure 13. The division of flow between forward and aft hoists and between port and starboard hoists is established by servo controlled flow dividers. The signal from the flow dividers is derived from a comparison of the signals from the rotary potentiometers mounted on each hoist. Two potentiometers are mounted on each of the starboard hoists, and only one will be mounted on each of the port hoists. The first potentiometer on the star-

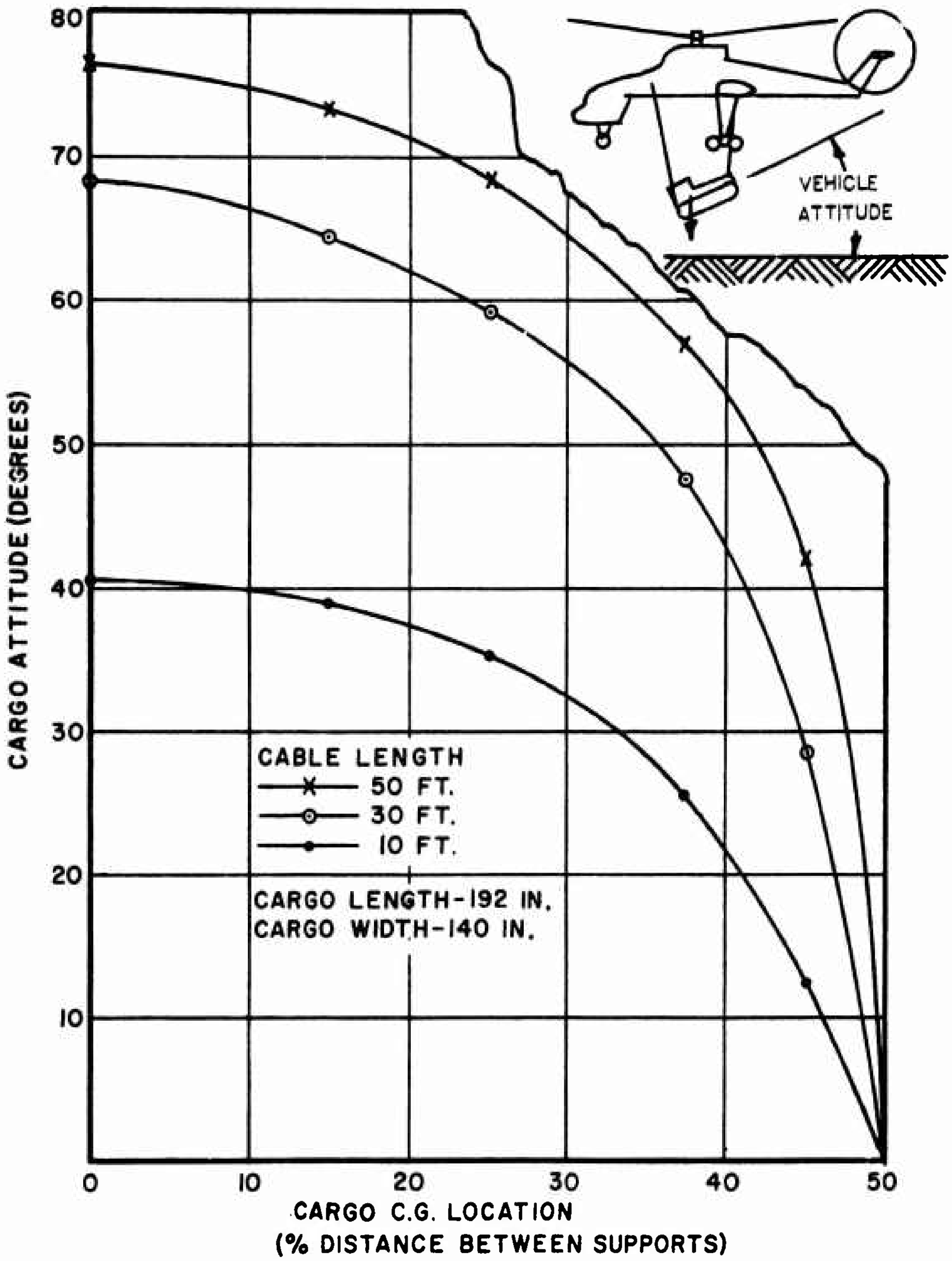


Figure 12. Multi-Point Hoist - Equalized Load System, Cargo Attitude vs C.G. Location.

board hoists controls the division of flow between the two forward and the two aft hoists. The second potentiometer on the port and starboard hoists controls the distribution of flow between the two port and the two starboard hoists. Operation of the hoists individually (beeping) is accomplished by supplying a bias signal to the two relevant flow dividers, blocking flow to three of the hoists.

Clutches disengage the feedback potentiometers from the hoists during beeping operation and reengage for collective operation. This enables the cables to be adjusted for any extremes in cargo shape and then permits the established cable lengths to be maintained during collective operation. This system eliminates errors due to differences in motor efficiency and initial settings of the divider valves. The maximum error is estimated to be in the order of 7-3/4 inches in 50 feet (see Error Analysis, page 148).

By deenergizing the feedback system during the initial stages of hoisting, the cable loads are automatically equalized, since the basic system is inherently load equalizing. Figure 12, page 46, shows the resultant load attitude with varying C.G. locations if an automatic load equalizing system is used.

Four-Point Hoist Control System - Mechanical Power Source

A clutch-reverser unit mounted on the accessory gearbox provides power to both the single- and four-point hoists. Clutches mounted on angle gearboxes adjacent to each of the hoists permit individual operation as required. A separate clutch is used to disengage the single-point hoist during operation in the four-point mode.

Synchronized operation is attained by engaging the four-point hoist clutches and then engaging the clutch-reverser unit. This operation is automatically sequenced so that only one control motion will be required.

Equal cable loading is attained by disengaging the clutch driving the heavily loaded hoist and allowing its load brake to maintain the load while operating the lightly loaded hoists until equal cable loading is attained. Load indicators will make it possible to determine when equal loading is obtained.

Since this system is awkward to use effectively, it would be desirable to incorporate a feedback system to give load equalization automatically. Such a system utilizes the output of the load cells to supply a control signal to the proper clutches. This signal permits clutch slippage until equalized loading is attained. A temperature sensor in the separate clutch units would provide protection against overheating by locking up the clutch to prevent excessive heat buildup.

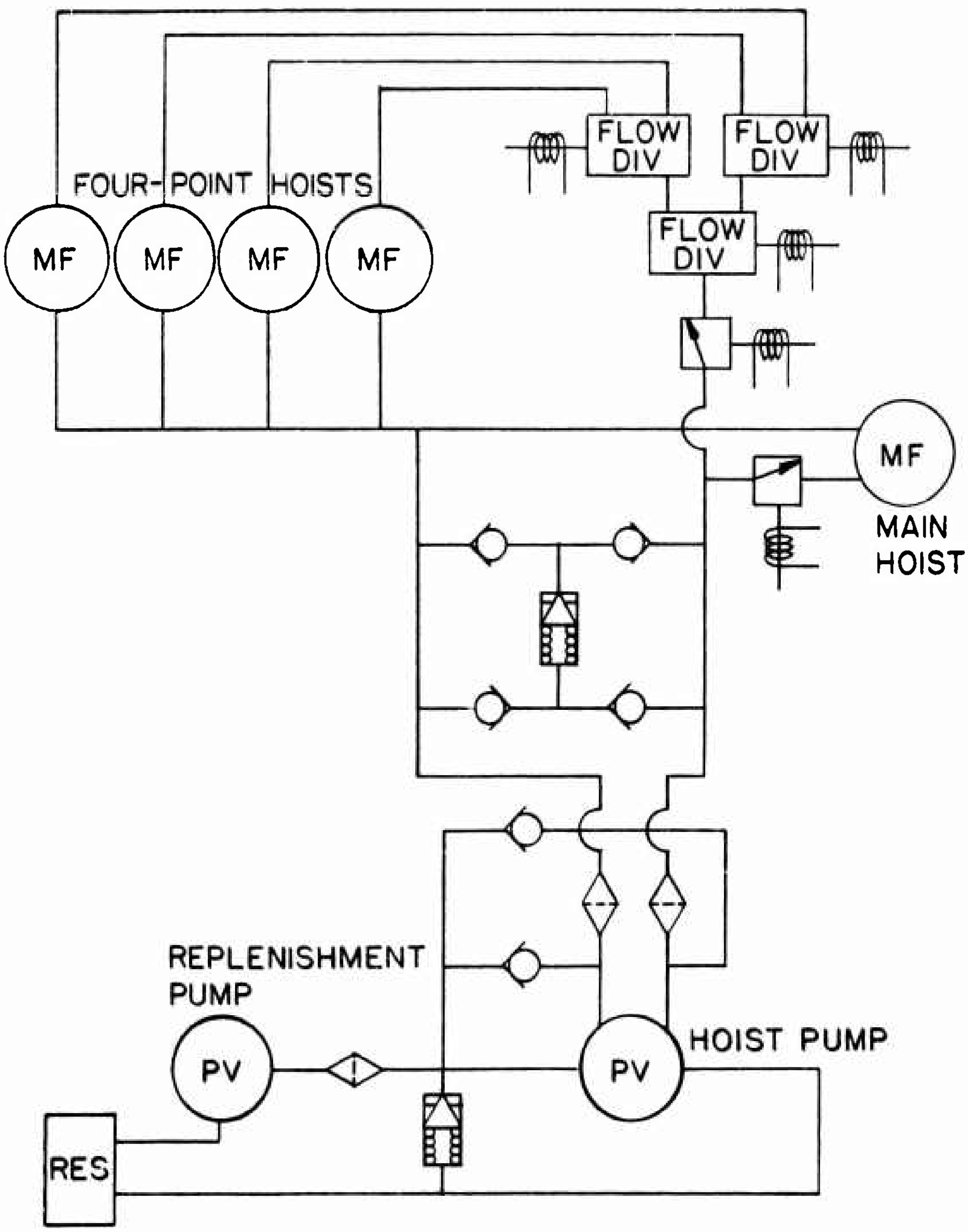


Figure 13. Hydraulic Schematic - Single-Plus Four-Point Suspension.

ISOLATORS

Isolators are required on all hoists to eliminate the vertical bounce phenomena (see page 107 for a discussion of vertical bounce).

The isolators are of the hydraulic cylinder type similar to that used on the CH-54A main cargo hoist. This type of isolator incorporates an isolator, a load cell, shock struts, and a charging cylinder in one unit. The shock struts serve to retard the return stroke of the isolator when loads are air dropped. The charging cylinder which is pressurized by the aircraft utility hydraulic system compensates for temperature induced pressure changes in the isolator and makes up for any leakage that may occur.

When applied to cargo hoists of conventional design, the isolator reacts the cable load through a linkage. On zero-moment hoists, it is mounted directly in line with the cable so that no linkages are required.

HOIST CABLES

Background

The main cargo hoist used on the CH-54A has one of the largest capacities in existence. It is capable of raising and lowering a 15,000-pound load at 45 feet per minute and has a static lift rating of 20,000 pounds. A 7/8 diameter (.923 actual) nonrotating cable of 18 x 7 construction with individual wires .058 inch in diameter is used to support the load. The core of the cable contains seven electrical conductors wrapped in a resilient Teflon jacket. The electrical conductors are used to operate the hook indicator lights and to power a solenoid which opens the hook. The cable has a minimum breaking strength of 58,000 pounds. The cable is wrapped on a drum whose basic pitch diameter is 24.5 inches, which gives a drum to cable wire diameter ratio of 422 to 1. The drum is magnesium to which a 0.25-inch-thick polyurethane rubber jacket is molded. The rubber jacket serves to reduce chafing, thus prolonging both drum and cable life. Experience with this hoist in the Southeast Asian theatre of operations for 11 months has not resulted in a single cable fatigue failure.

Single-Point Hoist Cable Design

To meet the 40,000-pound single-point hoist capability of the H.L.H., it will be necessary to use a cable diameter of 1.39 inches. An 18 x 19 construction is used to obtain a flexibility greater than that of the 18 x 7 construction used on the CH-54A.

Normal nonrotating cable construction requires the use of 18 strands of wire with either 7, 19, or 37 wires in each strand. Therefore, cable flexibility is increased by using a larger number of smaller diameter wires in each strand rather than by increasing the number of strands. The 18 x 19 construction cable uses a wire diameter of 0.060 inch compared to a

wire diameter of 0.058 inch in the 18 x 7 cable. Using commercial practice (for cables without conductors in the core) which requires a cable drum diameter 400 times the individual wire diameter, it would be possible to use a cable drum diameter of 24 inches. Use of 18 x 7 construction cable with 7 wires of .085 diameter per strand would, by the 400 to 1 rule, require a minimum cable drum diameter of 34 inches.

While the drum diameter selected is greater than the 24 inch minimum allowable, the extra flexible construction has been retained, since it increases the fatigue life of the cable. The determining factor in selection of the drum diameter is the requirement to carry as much cable as possible without exceeding desirable aircraft control limitations (see Single-Point Hoist section, page 17) rather than the minimum ratio of cable drum to wire diameter (the 400 to 1 rule).

Multi-Point Hoist Cable Design

For the 23,100-pound-capacity two-point hoists, a 1.06 diameter cable is required, and a 0.79 diameter cable is required for the 11,550-pound-capacity four-point hoists. All cables are stainless steel and are of 18 x 19 nonrotating construction. Wire size for 1.06 diameter cable is .047 and is .035 for the .79 diameter cable. The use of nonrotating construction in all of the multi-point hoists is desirable,since it will permit them to carry separate cargo on individual hoists. The four hoists, for example, could be rigged to carry separate fuel bags, or sling loads of ammunition, to individual sites. All cables will contain seven electrical conductors suitably protected by a resilient Teflon jacket in the central core. This construction provides the maximum protection for the conductors from both the elements and from damage due to rough handling.

No development problems are expected in the fabrication of any of the three cables described above.

The one major development problem to be solved is that of providing a cockpit controlled mechanical release of the load that can be integrated with the load suspension cables. Several proposed solutions to this problem and an alternate method of providing the needed redundancy in release methods are described in the PROBLEM AREAS AND PROPOSED SOLUTIONS section, page 113.

CARGO HOOKS

Background

Cargo hooks used by helicopters for support of external loads have undergone extensive development since the original manually released hooks were first introduced. Capacities have increased from 2000 pounds to the 20,000 pound capacity hook presently used in both the CH-53A and the CH-54A. Electrical release modes have been added and the automatic touchdown release has been developed. Indicator lights for load beam attitude also have been added.

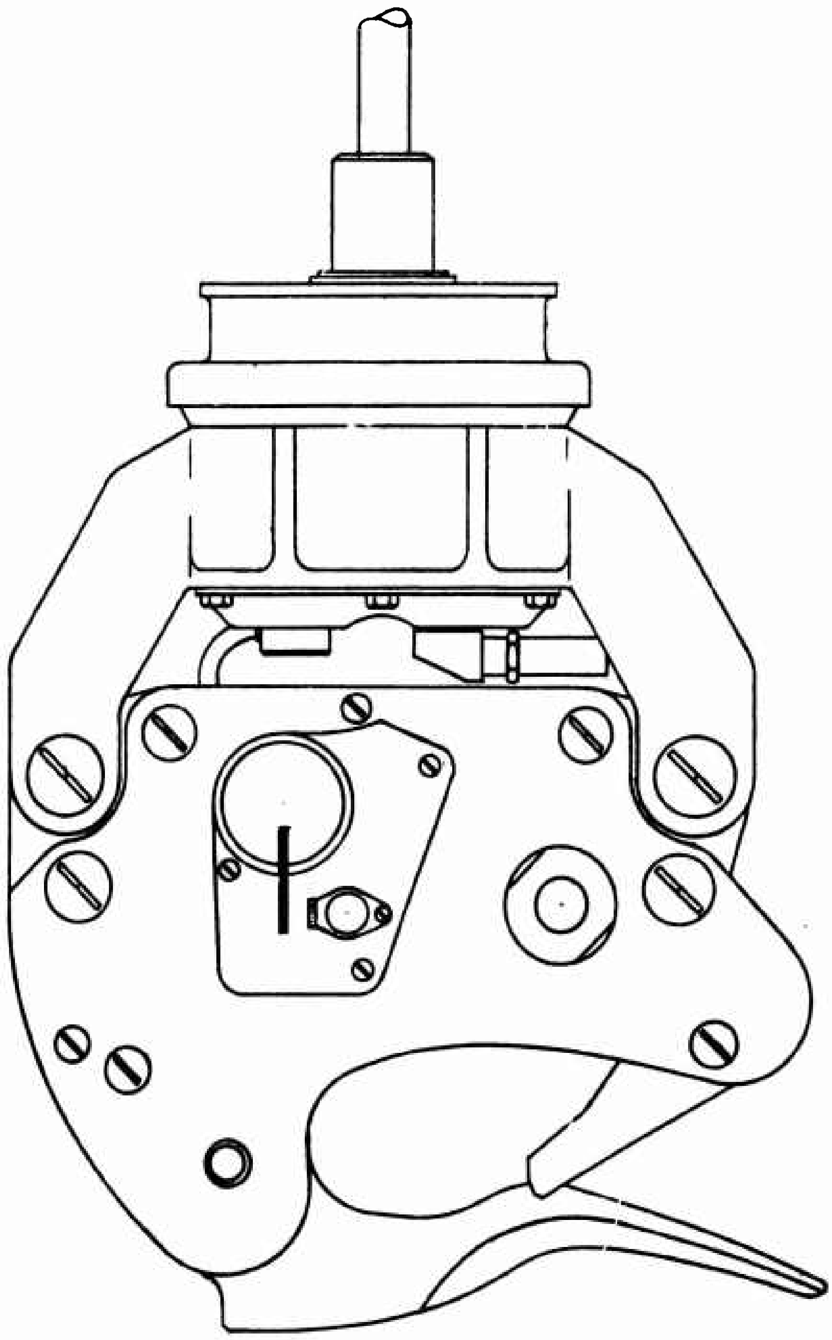


Figure 14. CH-54A Hook - Swivel Assembly.

Table XV shows the weight trend of cargo hooks presently in use. It should be noted that the hooks rated at 20,000 vary in weight by 27.2 pounds, with the hoist mounted hook being heavier. This increase in weight is due solely to the requirement for a swivel-slip ring assembly and not for other reasons such as cable straightening requirements.

Heavy Lift Helicopter Cargo Hook Design

To meet the cargo hooks requirements for hoists of this study, the basic data shown in Table XVI, page 53, were generated and design proposals were solicited from several manufacturers of airborne cargo hooks.

TABLE XV
HELICOPTER CARGO HOOKS

Aircraft	Type of Suspension	Normal Operating Load (lb)	Ultimate Load (lb)	Weight (lb)	Comments
H-34	Sling	5,000	25,000	10.7	
H-37	Sling	10,000	50,000	27	Requires manual relatch
CH-3C	Sling	10,000	50,000	24	
CH-53A	Pipe	20,000	90,000	40	
CH-54A	Hoist	20,000	90,000	67.2	Requires swivel-slip ring assy

Proposals received from hook manufacturers indicate that all these requirements can be met with no major advances being required in the state of the art. The weight of the 40,000-pound-capacity hook swivel assembly will be 192 pounds, that of the 23,100-pound unit will be 87 pounds, and the 11,550-pound unit will be 59 pounds. All units will be similar in design to the assembly presently in use on the CH-54A, shown in Figure 14, page 51. It is the opinion of one hook manufacturer that the weight of the 40,000-pound-capacity unit can be significantly reduced by a change in the relationship of the structural parts. Such a change will be actively pursued in Phase II of this study.

Field experience with the CH-54A has borne out the fact that the swivel-slip ring assembly is the most sensitive part of the total hook assembly to environmental conditions. Such assemblies must be designed to provide the utmost protection from the elements. In addition, they must be rugged enough to withstand repeated abuse. Environmental testing is mandatory to ensure that the required protection has been provided. A refinement, currently being studied under a separate contract, is the incorporation of an AN 4064 type dehydrator for both the hook and swivel-slip ring assemblies.

TABLE XVI
BASIC DATA - CARGO HOOKS

GENERAL DESIGN REQUIREMENTS

1. Open throat design of hook
2. Automatic relatch of load beam
3. No safety lock
4. Both manual and electrical release modes required
5. Swivel-slip ring assembly required to allow hook rotation*
6. Slip ring to have 7 circuits
7. Design life: 5000 full load releases
8. Indicator signals for hook open and hook closed
9. Environment: -65°F to 130°F, sand and dust, fungus, water immersion
10. Hook detachable from swivel; swivel detachable from cable
11. Hook supported by a single cable with hollow core for electrical and/or mechanical conductors.

Capacity	A	B	C
Cable Size	1.39	1.06	0.79
Max. Operating Load	40,000	23,100	11,550
Limit Load	100,000	64,700	32,350
Ultimate Load	150,000	97,000	48,500

*Swivel assembly, combined with nonrotating cable construction, permits individual loads to be carried on multi-point hoist systems (see page 50).

This will provide field-level personnel with a method to check for moisture contamination without disassembly.

MISCELLANEOUS COMPONENTS

Clutch Design

All hoist distribution clutches are of the multiple-disc, wet-plate type. They will be either hydraulically or electrically actuated. They are spring loaded to the disengaged position. The clutches are mounted on the angle gearboxes and are easily removable.

Traction Sheaves

All traction sheaves will be hydraulically powered and universally mounted. The power required to drive the sheaves was established by assuming a minimum cable sag of 1 inch in 19 inches of span and a requirement that the sheave operate at a speed 5 pct above that of the hoist during the lowering mode of operation. Driving friction of the cable in the sheave is assured by the use of adjustable pressure rollers. A slip clutch is used in series with the motor. The clutch will be immersed in oil to permit proper cooling during operation.

The sheave will be removable; both the cable cutters and the bellmouth will be slotted to permit installation and removal of the cable. Tandem-dual cable cutters are mounted on the bellmouth. The use of pressure rollers and a cover for the sheave provides an effective backlash suppressor in the event that the cables must be sheared. A typical design is shown in Figure 15, page 55.

Cable Cutters

All cable cutters are of the tandem-dual type presently installed on the four-point hoists being developed for use on the CH-54A.

Two cable cutters of the electrically ignited, explosively propelled knife type are mounted in tandem on both hoist and traction sheave bellmouths. Each cutter's explosive charge can be fired by either of the two bridge-wire circuits. Separate routing of the wiring for each of the circuits provides additional redundancy. All firing circuits are actuated simultaneously. Attachment of the cable cutters to the bellmouth will permit easy tear-out in the event that the lower cutter should trap the cable in the housing instead of shearing it.

All hoists incorporate the tandem-dual cable cutter concept to meet emergency release requirements. Since the circuitry required to fire the cable cutters bypasses the hoist and hook slip ring assemblies as well as the load suspension cable, this concept meets all requirements for a redundant release system. All wiring to the cutters will be installed with adequate slack to allow free hoist movement and will be armored to protect from accidental damage.

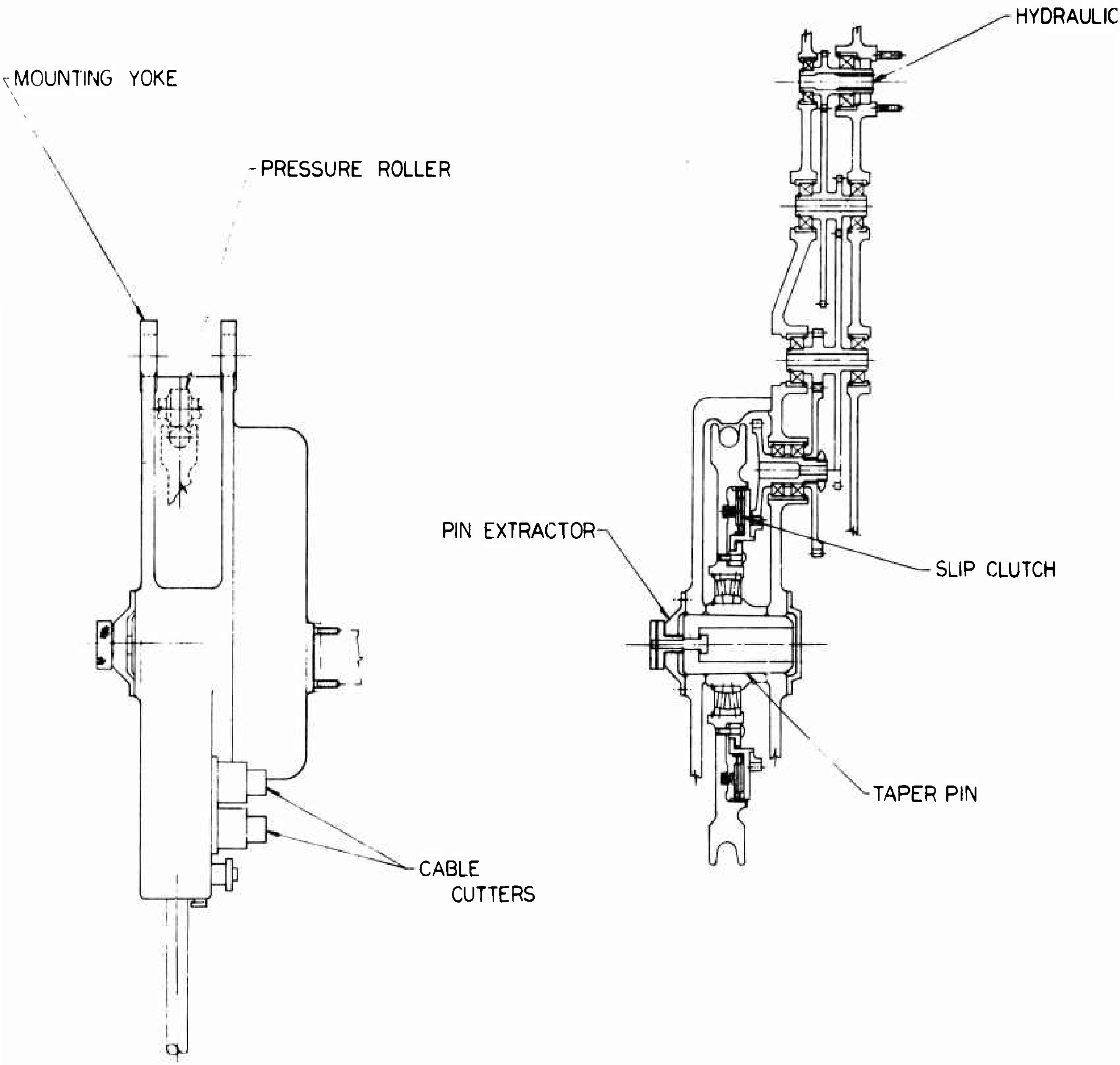
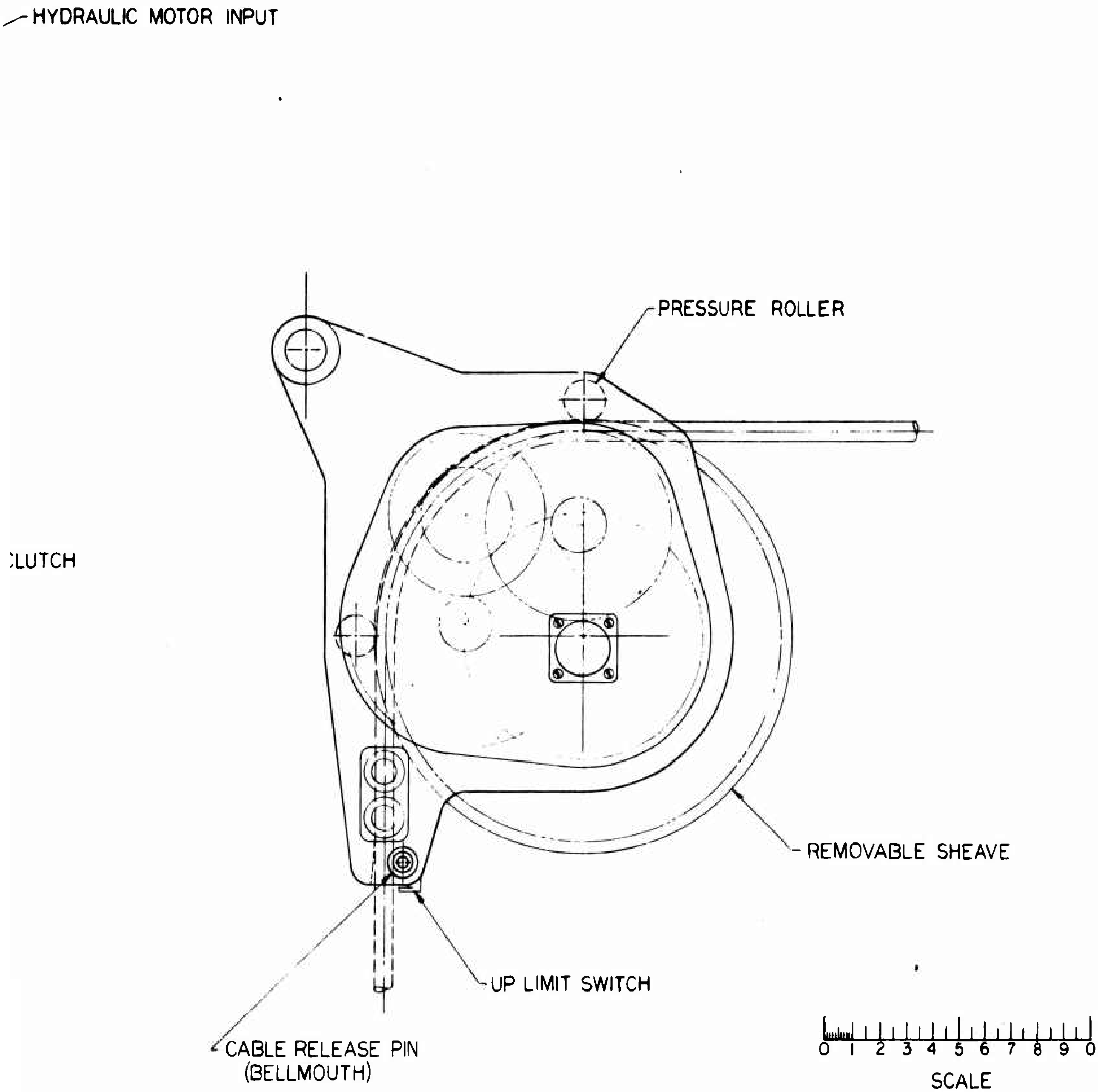


Figure 15. Traction Sheave.

A.


HYDRAULIC MOTOR INPUT
LUTCH
PRESSURE ROLLER
REMOVABLE SHEAVE
UP LIMIT SWITCH
CABLE RELEASE PIN
(BELLMOUTH)
0 1 2 3 4 5 6 7 8 9 0
SCALE

B.

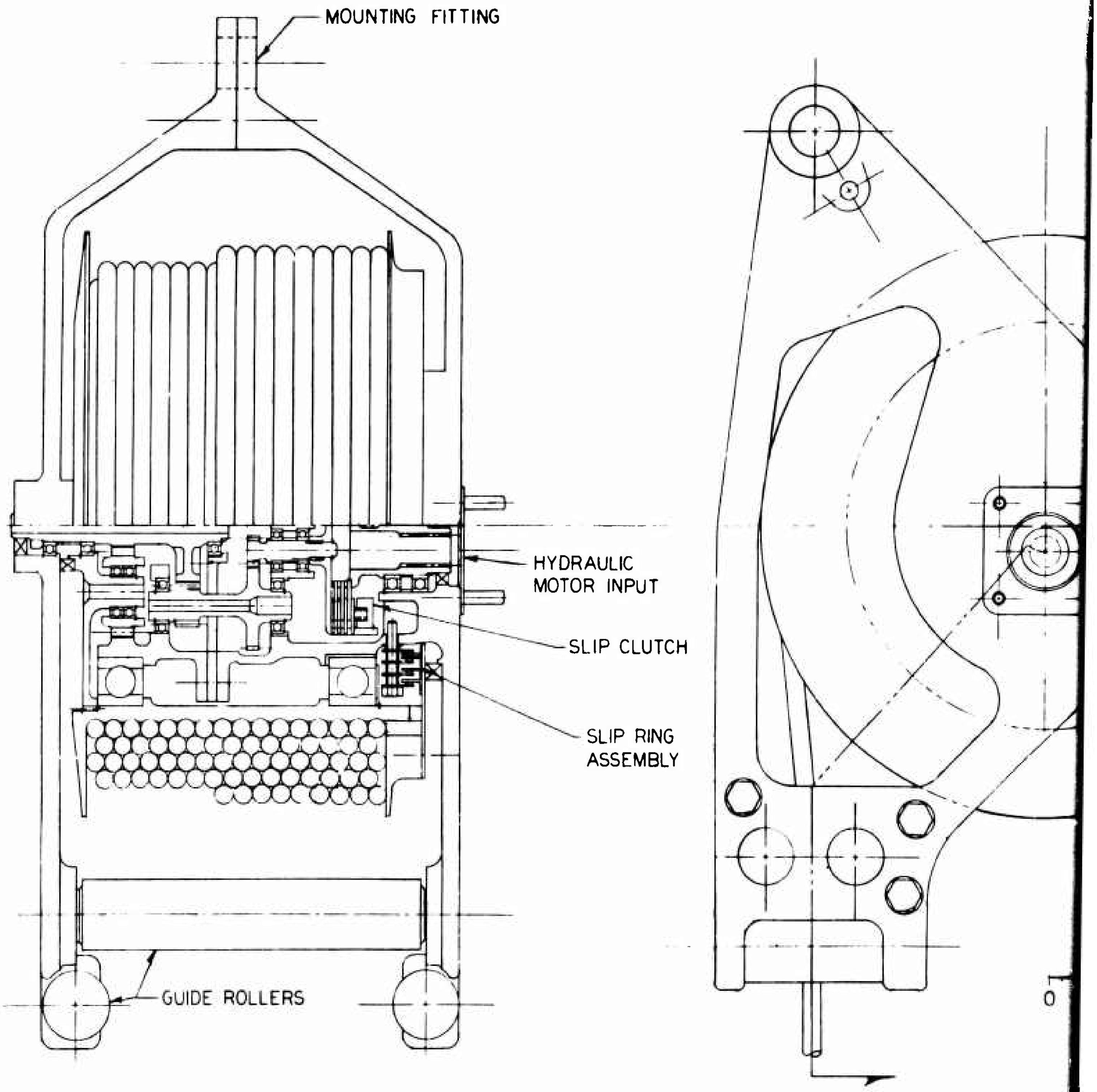
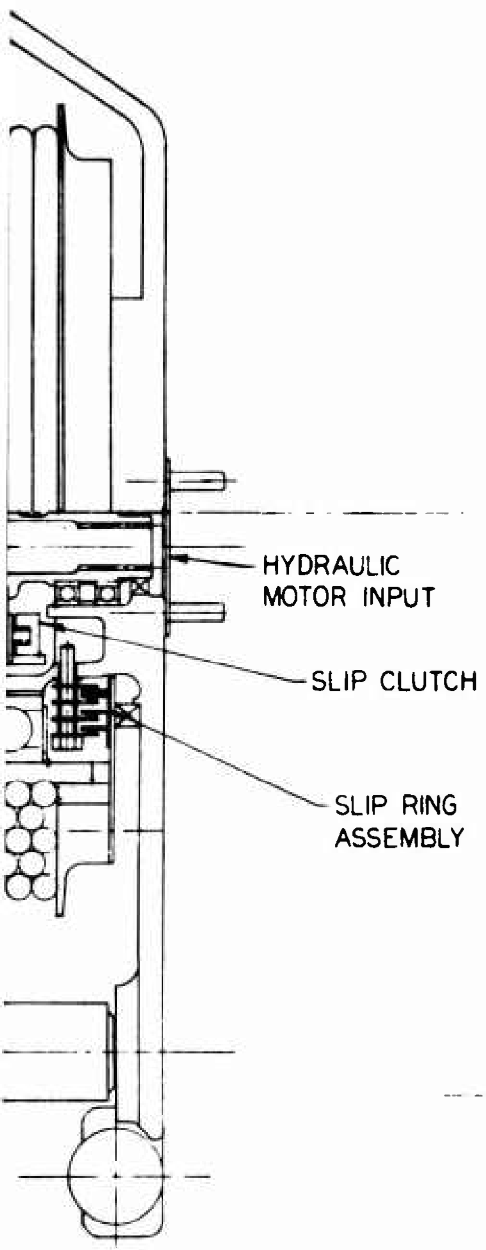
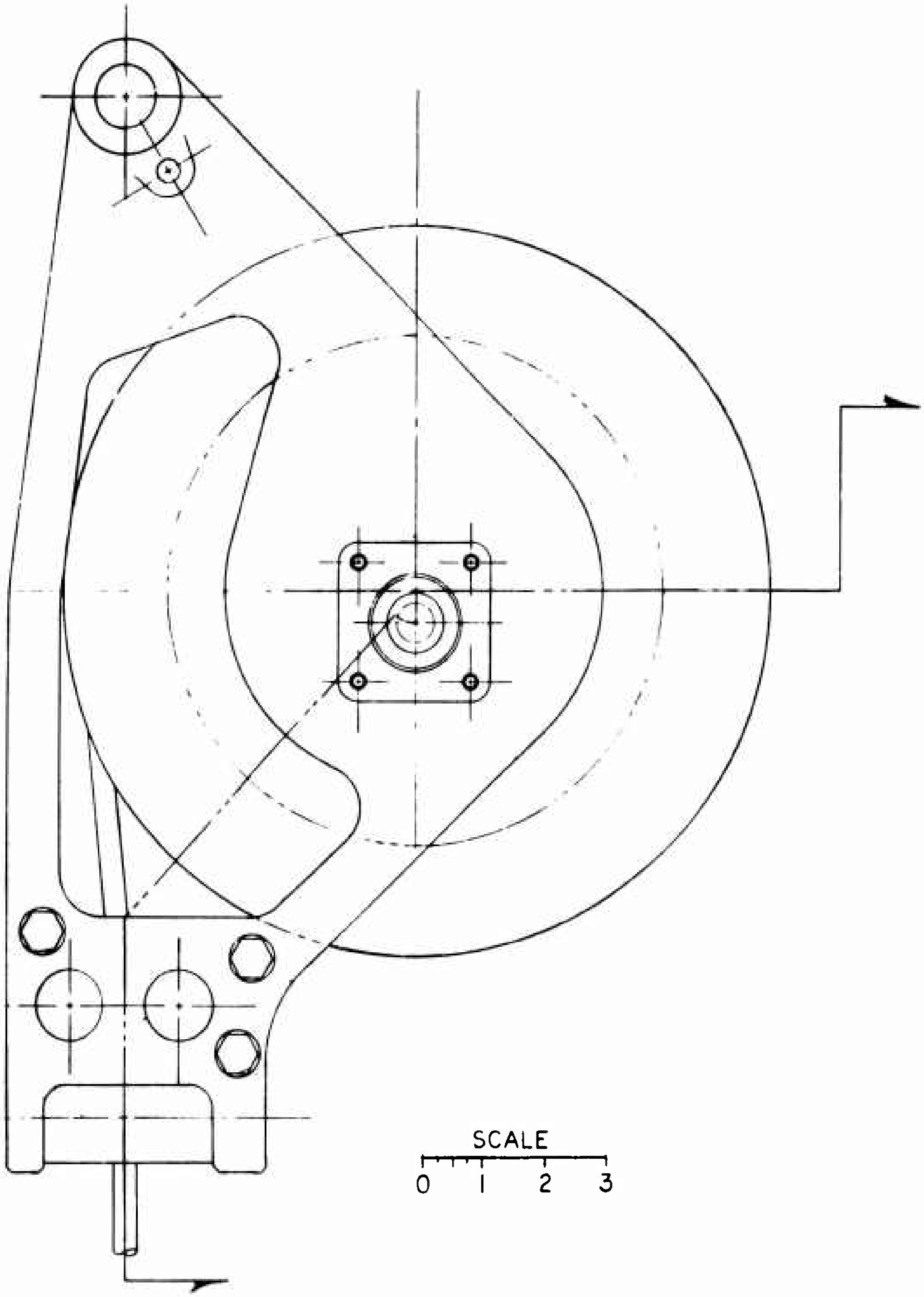


Figure 16. Conductor Reel.

A.

JNTING FITTING





ctor Reel.

A.

B.

The "all-fire" current will not exceed 16 amperes per hoist. Thus the emergency release system can be operated on battery power only in the event of the failure of both aircraft generators.

Conductor Reels

A conductor reel is required in several of the combined function configurations investigated to provide a method of supplying electrical power to the 40,000-pound-capacity cargo hook. In addition, the conductor reel is used to stow the hook out of the way when it is not in use.

The basic design will consist of a hydraulically powered hoist capable of storing 150 feet of electrical conduit. The conduit will have 7 electrical conductors in the core and will be suitably protected by a braided wire jacket. A slip ring assembly will transmit electrical power from the aircraft system to the electrical conduit. The use of a flat coil spring to provide power for reeling in was not considered feasible because of the 150 foot length of cable required. A 1.0-horsepower hydraulic motor driving through a slip clutch will permit the hook to be lowered without slack.

The motor, which will be synchronized to operate with the hoists, will also operate at a speed such that no slack will be permitted in the cable during lifting. The slip clutch will prevent the electrical conduit from lifting more than the hook weight alone and will permit the hook to be lifted into a storage well when not in use.

The conduit will be attached by means of its steel braided outer jacket to the swivel assembly of the hook with a suitable electrical connector to provide electrical power.

While such a unit is not commercially available at the present time, design and fabrication will present no major technical problems. Its weight should not exceed 50 pounds. A typical design is shown in Figure 16, page 57.

HOIST SYSTEM CONFIGURATIONS

Thirteen basic hoist system configurations have been investigated to meet the heavy lift helicopter external cargo handling requirements. Seven configurations are designed to meet the single- plus four-point requirements and six are designed to meet the single- plus two-point requirements. Three variants of each type are combined function arrangements with the single-point function being replaced by combined operation of the multipoint hoists. Tables XVII and XVIII, pages 87 and 89, include a brief description and summary of pertinent data on each system. Schematic drawings of all the systems are presented in Figures 17 through 29, pages 61 through 85. The weights given are based on 80 feet of cable for the single-point hoist.

-1 CONFIGURATION

General Description

Single-point hoist mechanically driven. Hydraulically powered zero-moment hoists for the four-point system.

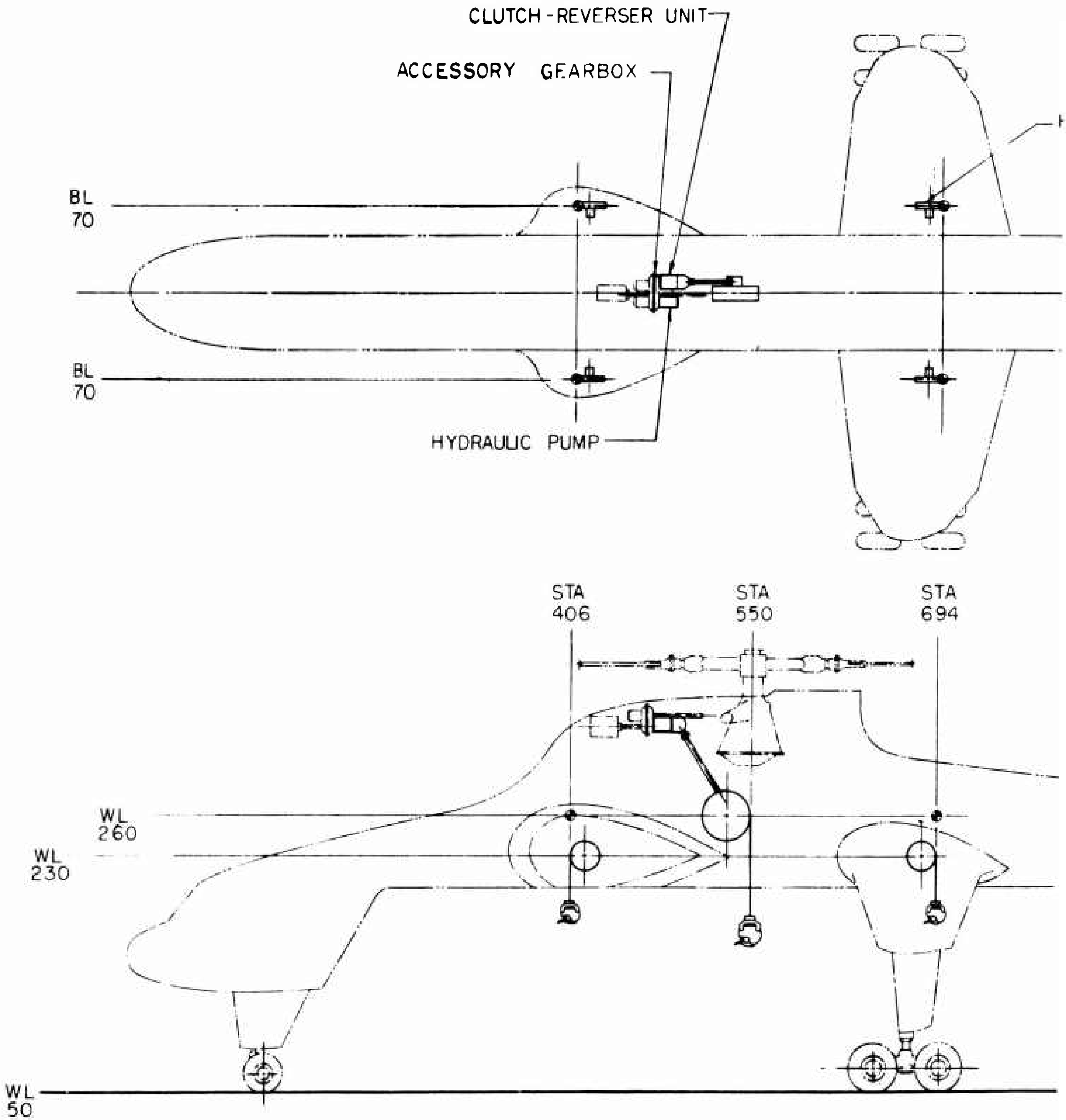
System Components and Weights

Single-Point Hoist (Type A)	1960 Pounds
Clutch-Reverser Unit	124
Angle Gearboxes (2 required)	43
Drive Shafts	6
Hoist Pump	35
Four-Point Hoist (Type K, 4 required)	2000
Hydraulic Motor (4 required)	40
Plumbing and oil	188
Total System Weight	4396 Pounds
Single-Point Mission Weight (Remove four-point hoists)	2396 Pounds
Multi-Point Mission Weight (Remove single-point hoist)	2436 Pounds

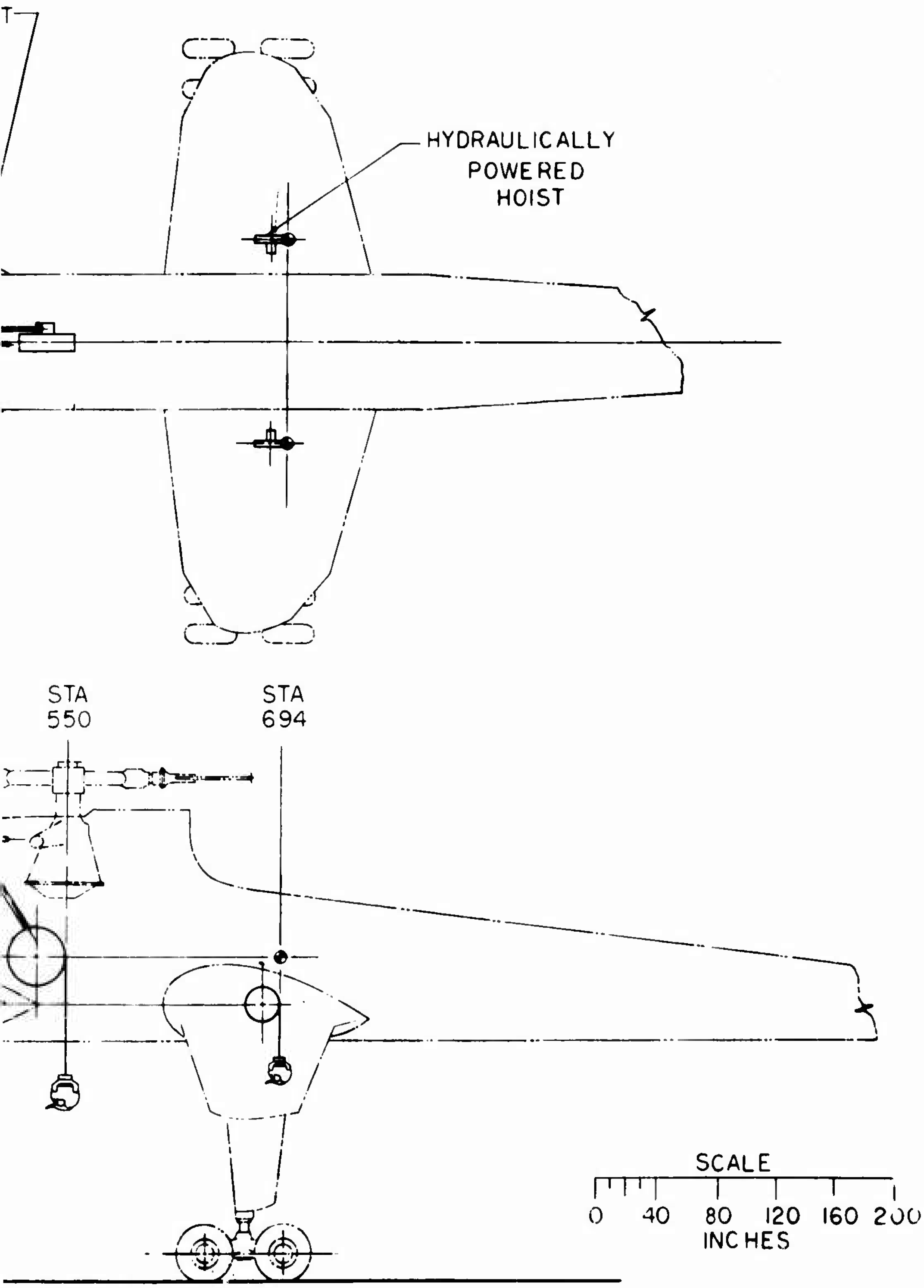
Power Required

Single-Point Mission	94.8 HP
Multi-Point Mission	84.8 HP

Figure 17. -1 Configuration.


CLUTCH-REVERSER UNIT
ACCESSORY GEARBOX
BL 70
BL 70
HYDRAULIC PUMP
STA 406
STA 550
STA 694
WL 260
WL 230
WL 50

A.



HYDRAULICALLY POWERED HOIST
STA 550
STA 694
SCALE
0 40 80 120 160 200
INCHES

B.

-2 CONFIGURATION

General Description

Single-point hoist hydraulically powered by one or two motors.
Hydraulically powered zero-moment hoists for the four-point system.

System Components and Weights

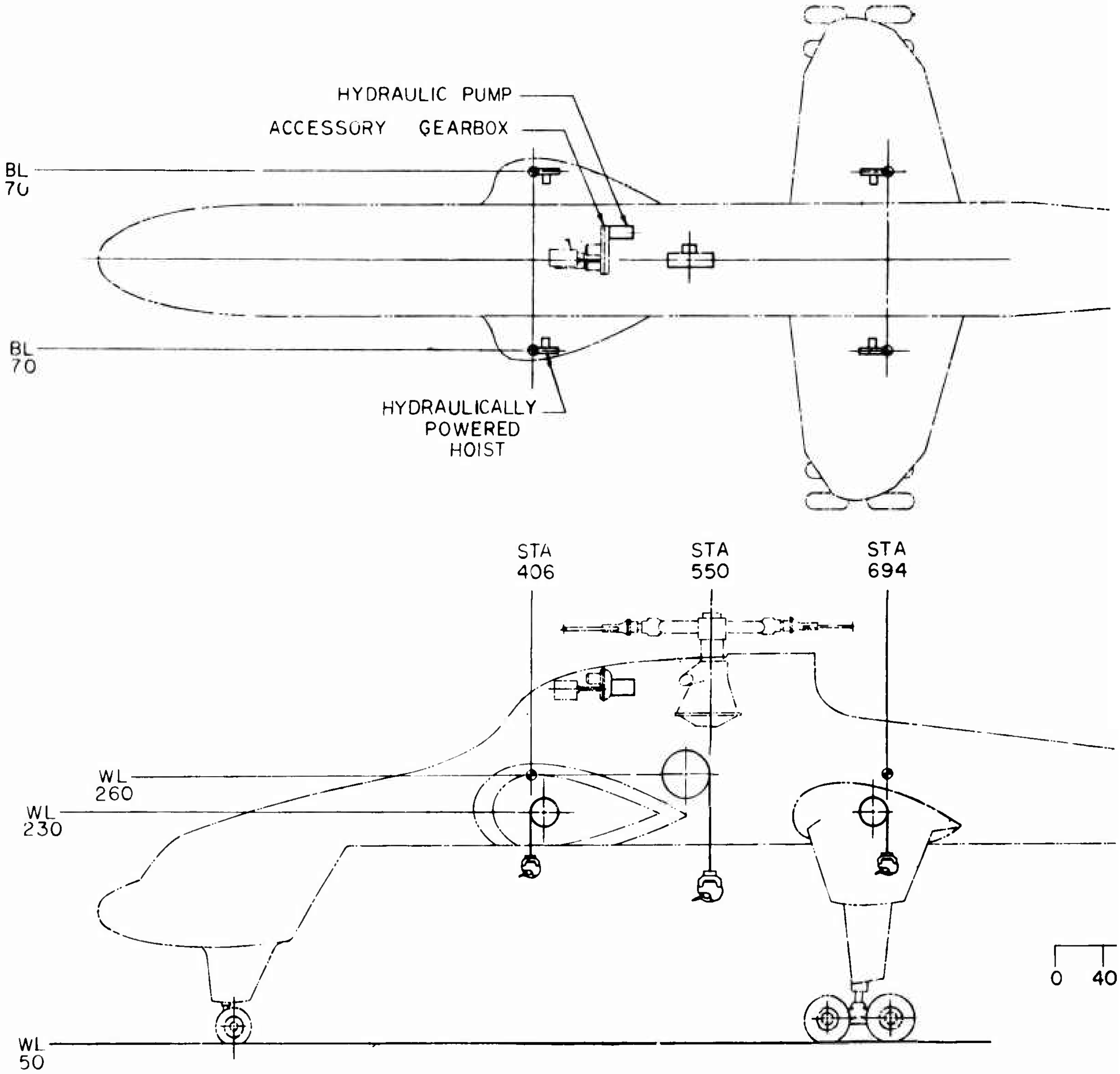
Single-Point Hoist	(Type A)	1960 Pounds
Four-Point Hoist	(Type K)	2000
Hydraulic Pump		55
Hydraulic Motors	(5 required)	84
Plumbing and Oil		188
Total System Weight		4287 Pounds
Single-Point Mission Weight (Remove four-point hoists)		2287 Pounds
Multi-Point Mission Weight (Remove single-point hoist		2327 Pounds

Power Required

Single-Point Mission	132.5 HP
Multi-Point Mission	84.8 HP

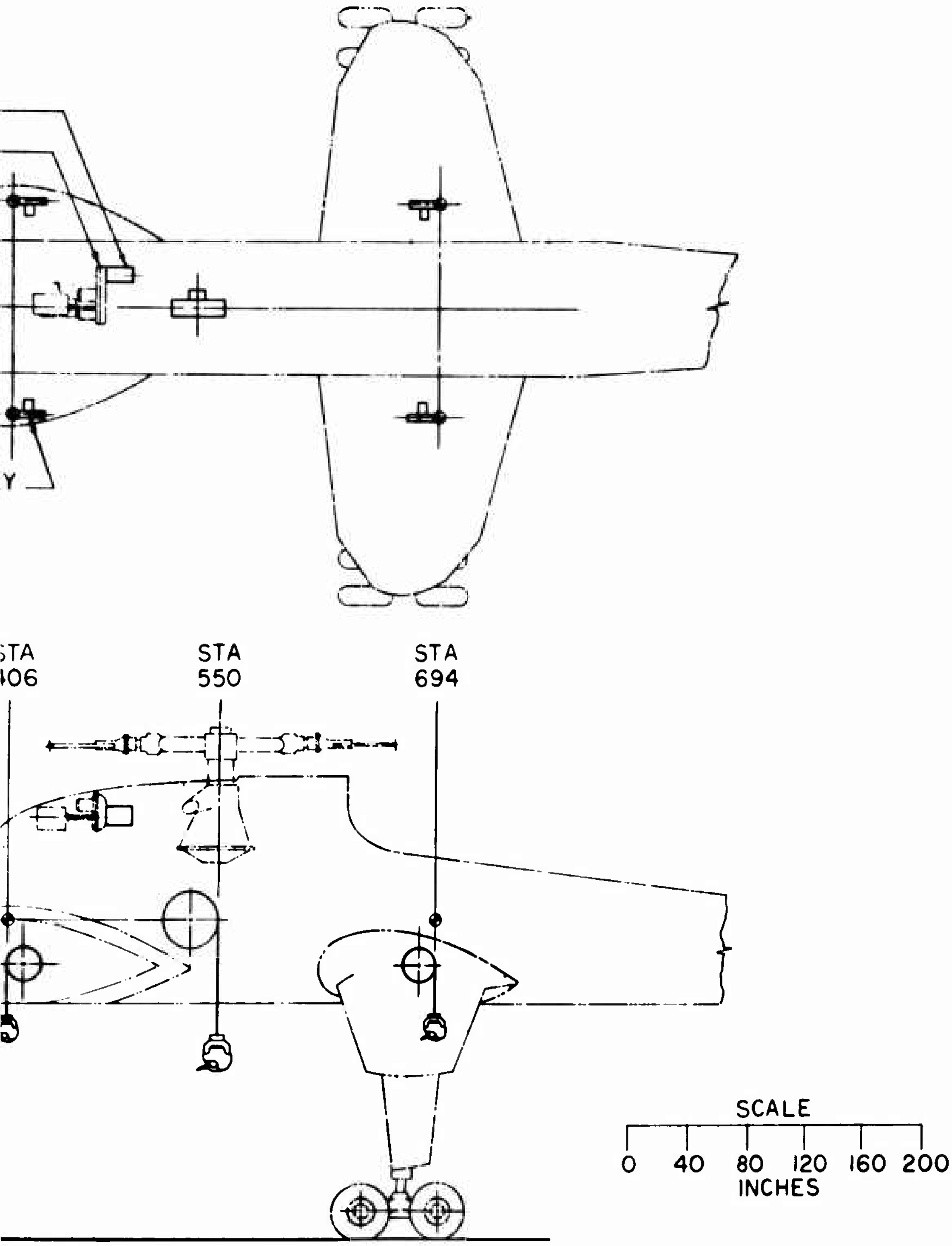
Figure 18. -2 Configuration.

A.



HYDRAULIC PUMP
ACCESSORY GEARBOX
BL
70
BL
70
HYDRAULICALLY
POWERED
HOIST
STA
406
STA
550
STA
694
WL
260
WL
230
WL
50
0
40

B.



e.

-3 CONFIGURATION

General Description

Single- and four-point hoists mechanically driven.

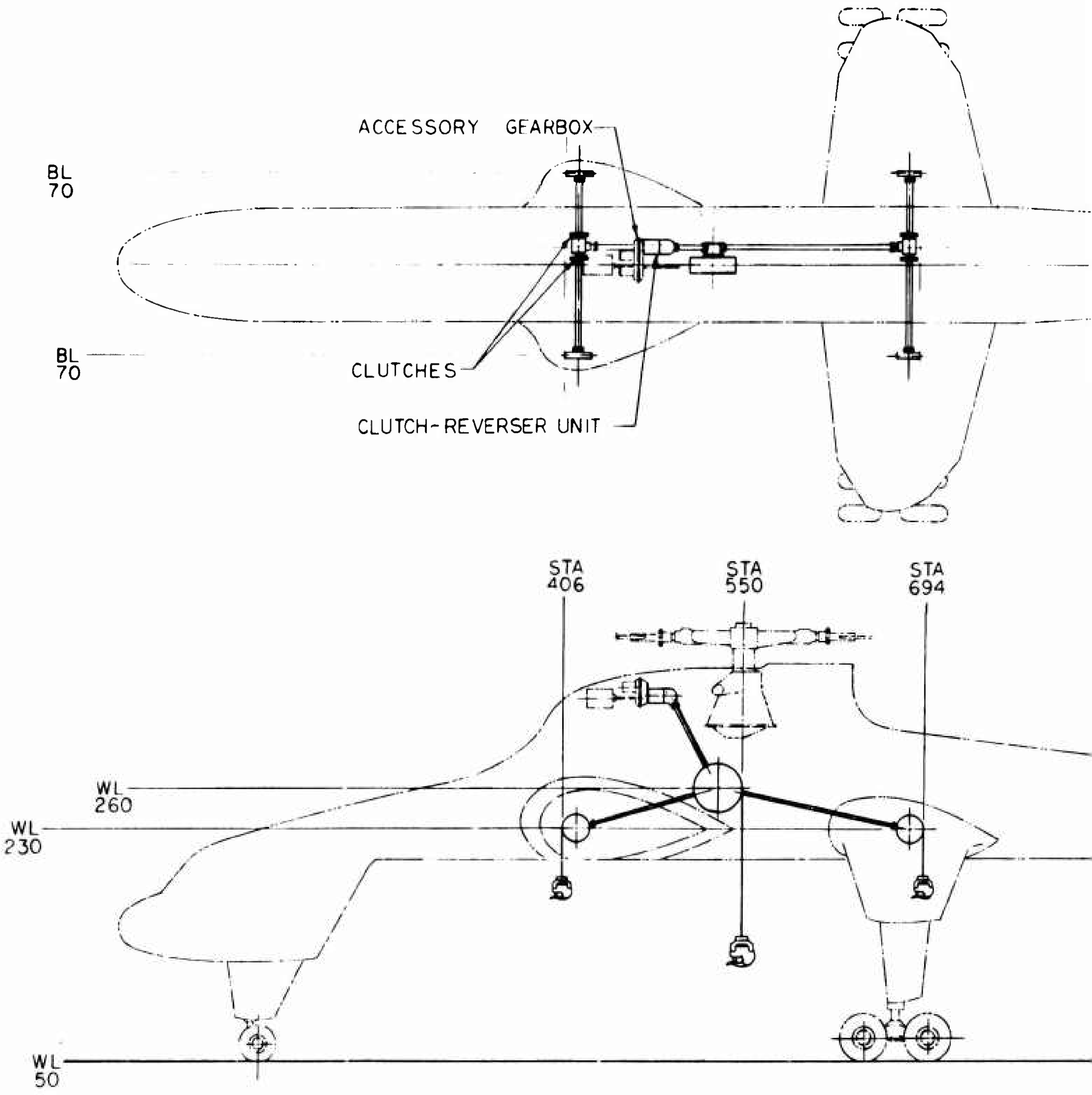
System Components and Weights

Single-Point Hoist (Type A)	1960 Pounds
Clutch-Reverser Unit	124
Angle Gearboxes (4 required)	80
Clutches (5 required)	51
Shafts and Couplings	45
Four-Point Hoists (Type P, 4 required)	1952
Total System Weight	4213 Pounds
Single-Point Mission Weight (Remove four-point hoists)	2261 Pounds
Multi-Point Mission Weight (Remove single-point hoist)	2253 Pounds

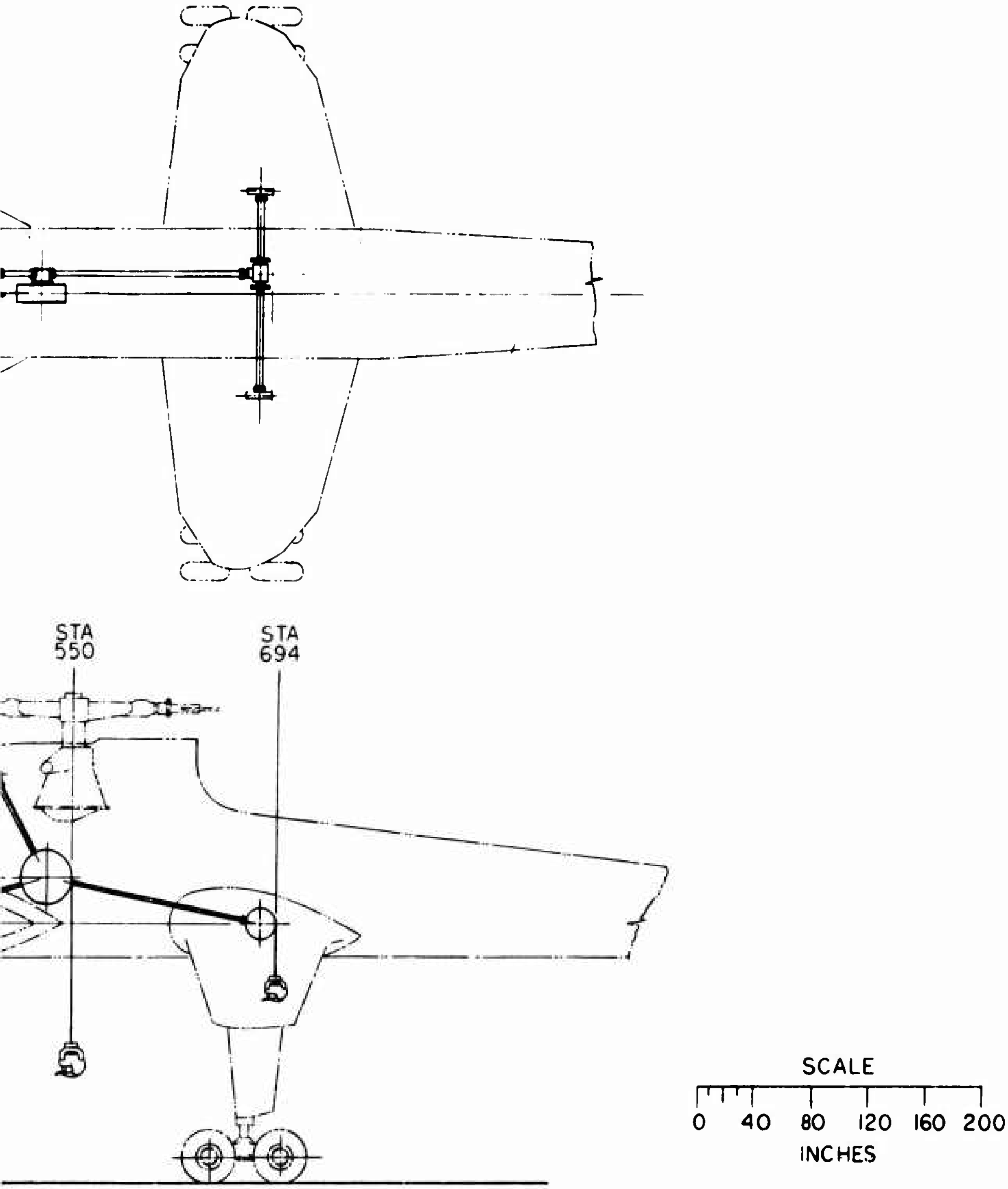
Powered Required

Single-Point Mission	94.2 HP
Multi-Point Mission	58.0 HP

Figure 19. -3 Configuration.


ACCESSORY GEARBOX
BL
70
BL
70
CLUTCHES
CLUTCH-REVERSER UNIT
STA
406
STA
550
STA
694
WL
260
WL
230
WL
50

A.



B.

-4 CONFIGURATION

General Description

Single-point hoist mechanically driven. Two mechanically driven, dual drum hoists, with cables reeved over hydraulically powered traction sheaves for the four-point system.

System Components and Weights

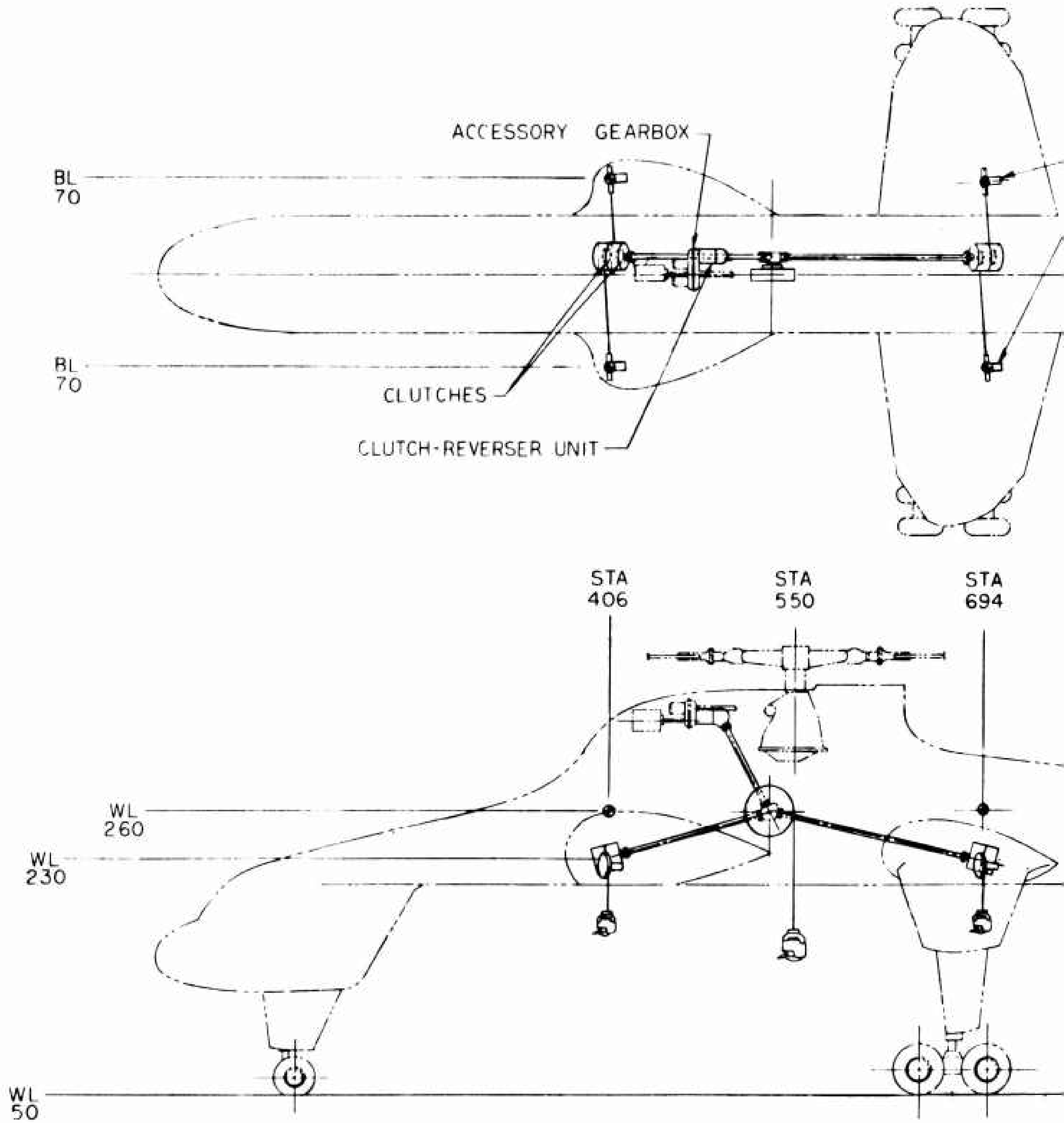
Single-Point Hoist (Type A)	1960 Pounds
Clutch-Reverser Unit	124
Angle Gearboxes (2 required)	42
Shafts and Couplings	29
Dual Drum Hoist (Type N, 2 required)	1980
Traction Sheaves	240
Clutches (5 required)	55
Total System Weight	4430 Pounds
Single-Point Mission Weight (Remove dual drum hoists)	2450 Pounds
Multi-Point Mission Weight (Remove single-point hoist)	2470 Pounds

Power Required

Single-Point Mission	94.1 HP
Multi-Point Mission	64.9 HP

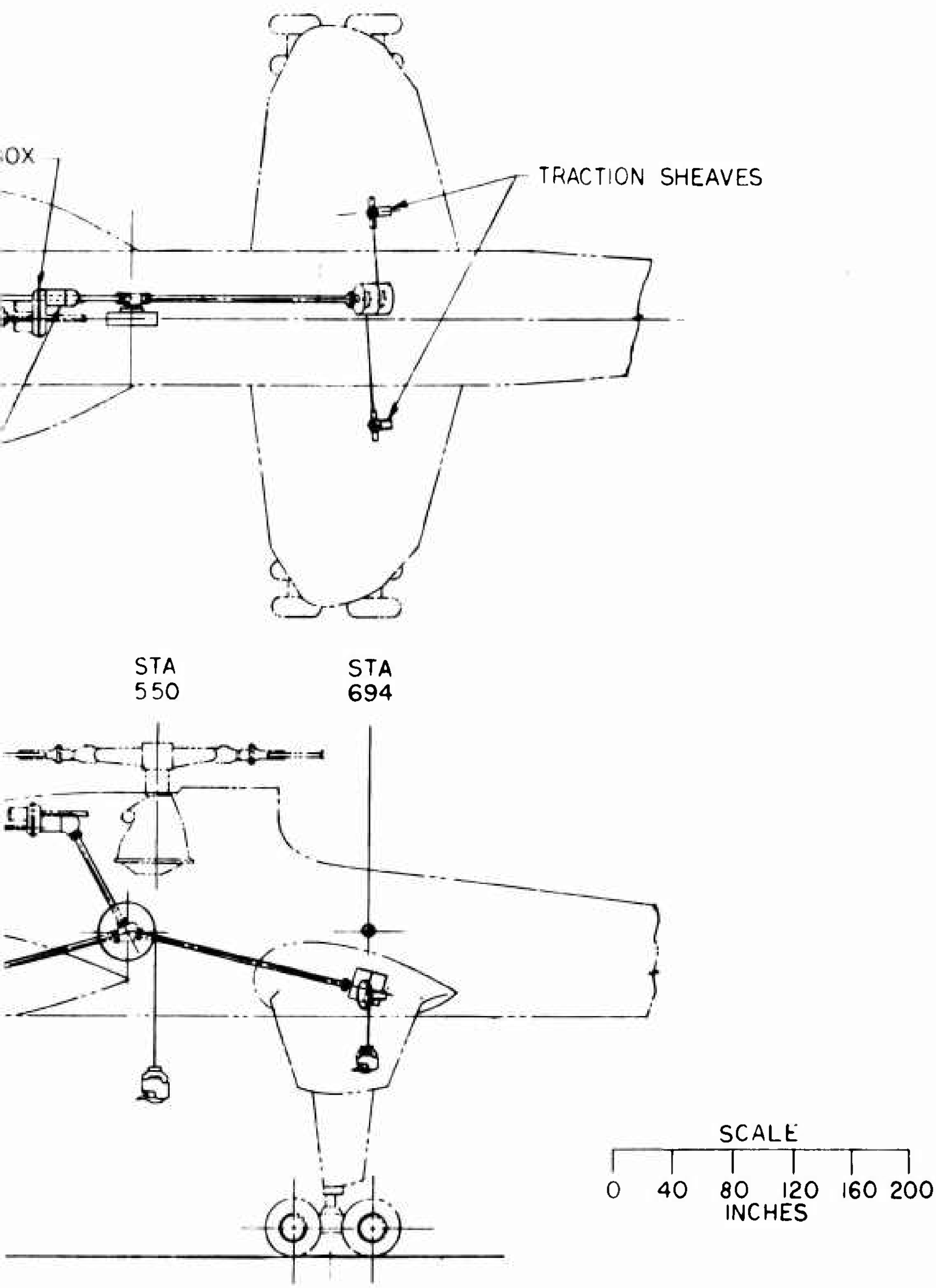
Figure 20. -4 Configuration.

n,



ACCESSORY GEARBOX
BL 70
BL 70
CLUTCHES
CLUTCH-REVERSER UNIT
STA 406
STA 550
STA 694
WL 260
WL 230
WL 50

A



OX
TRACTION SHEAVES
STA
550
STA
694
SCALE
0 40 80 120 160 200
INCHES

B,

-5 CONFIGURATION

General Description

No single-point hoist. Hydraulically powered zero-moment hoists for the four-point system with a frame, lockable to the aircraft to provide single-point capability.

System Components and Weights

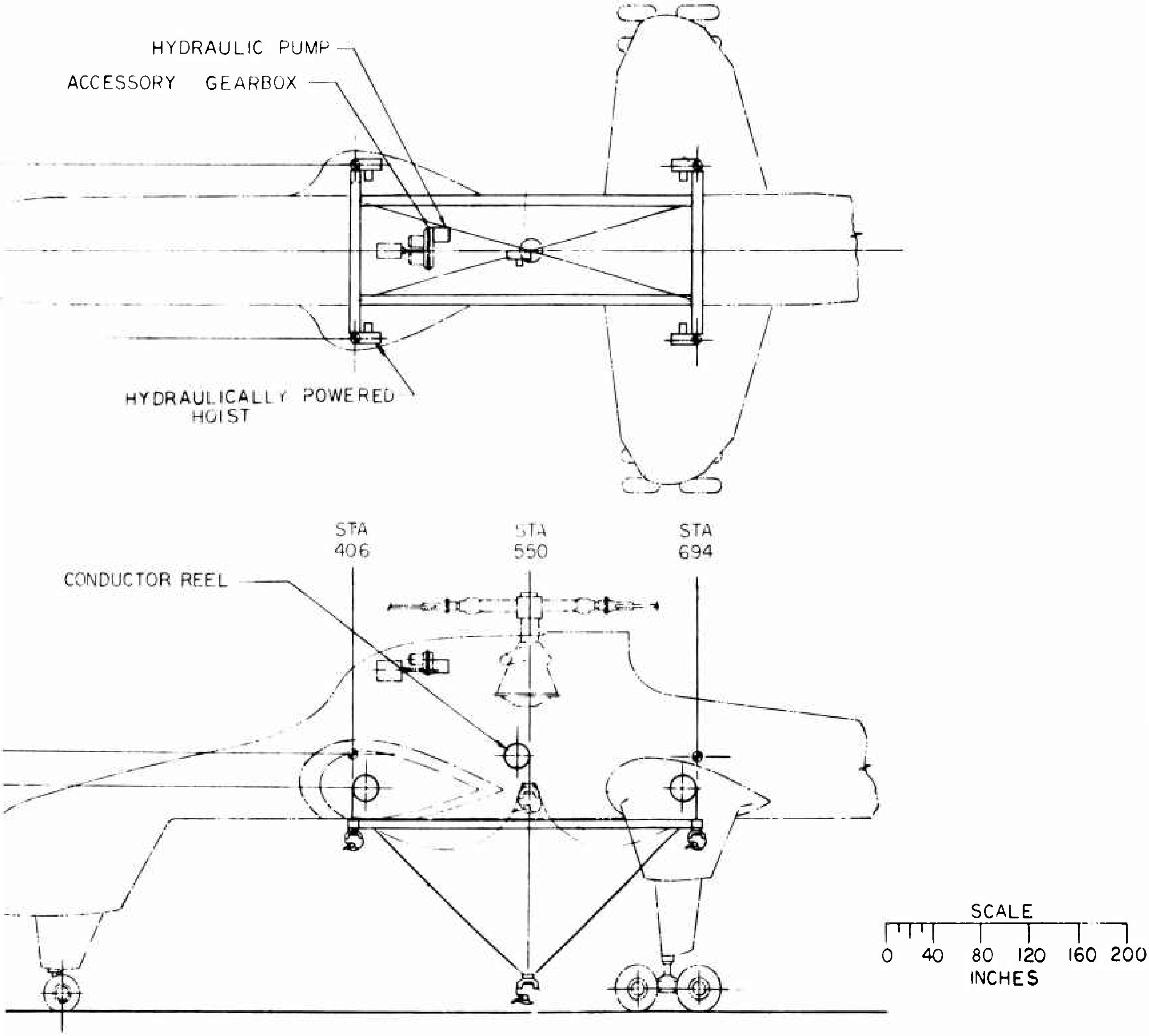
Four-Point Hoists (Type M, 4 required)	2200 Pounds
Hydraulic Pump	35
Frame (with slings & 40,000-lb hook)	1172
Plumbing and Oil	188
Hydraulic Motors (4 required)	88
Conductor Reel	50
Total System Weight	3733 Pounds
Single-Point Mission Weight	3733 Pounds
Multi-Point Mission Weight (Remove frame with slings & hook)	2561 Pounds

Power Required

Single-Point Mission	-
Multi-Point Mission	171.8 HP

Figure 21. -5 Configuration.

A\


HYDRAULIC PUMP
ACCESSORY GEARBOX
HYDRAULICALLY POWERED HOIST
STA 406
STA 550
STA 694
CONDUCTOR REEL
SCALE
0 40 80 120 160 200
INCHES

B1

-6 CONFIGURATION

General Description

No single-point hoist. Hydraulically powered zero-moment hoists with cables joined to a common hook for a single-point capability and reeved over hydraulically powered traction sheaves for the four-point system.

System Components and Weights

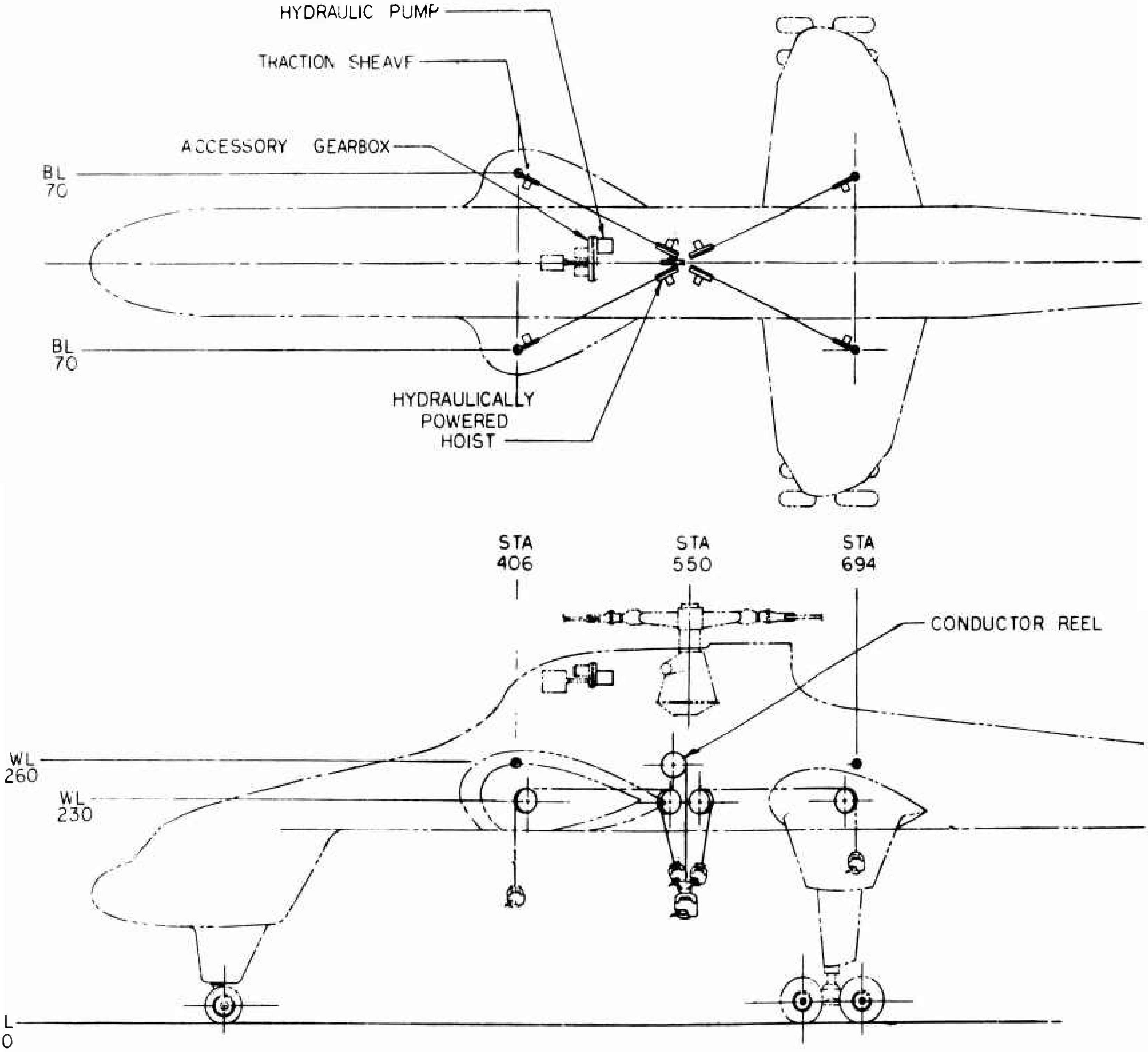
Four-Point Hoist (Type L, 4 required)	2200 Pounds
Hydraulic Pump	35
Traction Sheaves	240
Hydraulic Motors (4 required)	88
Conductor Reel	50
Hook and Swivel (40,000-lb Capacity)	190
Plumbing and Oil	188
Total System Weight	2991 Pounds
Single-Point Mission Weight	2991 Pounds
Multi-Point Mission Weight (Remove hook, swivel, & conductor reel)	2751 Pounds

Power Required

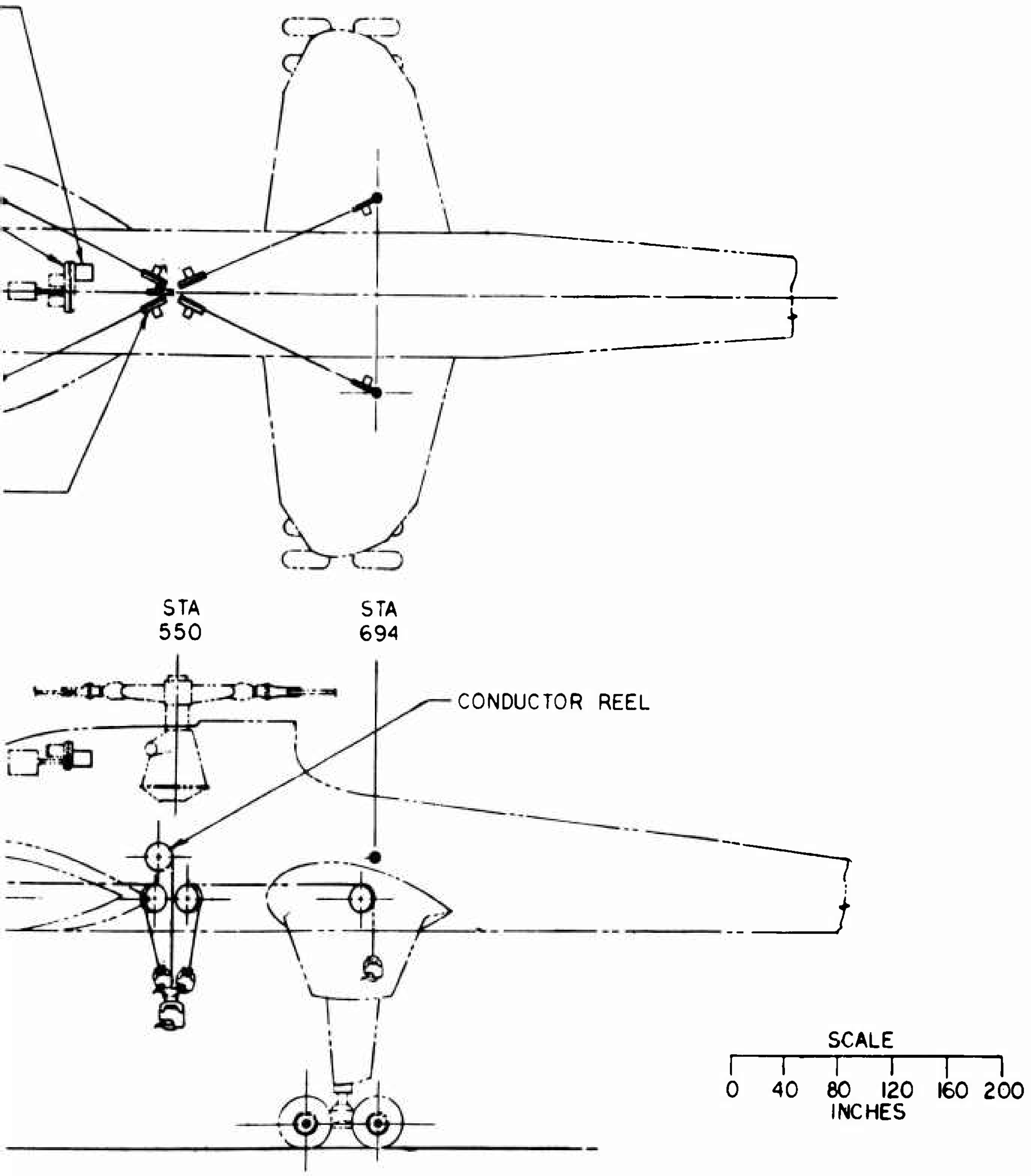
Single-Point Mission	-
Multi-Point Mission	179.1 HP

Figure 22. -6 Configuration.

A.


HYDRAULIC PUMP
TRACTION SHEAVE
ACCESSORY GEARBOX
BL 70
BL 70
HYDRAULICALLY POWERED HOIST
STA 406
STA 550
STA 694
CONDUCTOR REEL
WL 260
WL 230
L 0

B.



STA
550
STA
694
CONDUCTOR REEL
SCALE
0
40
80
120
160
200
INCHES

C.

-7 CONFIGURATION

General Description

No single-point hoist. Hydraulically powered zero-moment hoists for the four-point system. Cables joined by a master hook carried by one of the four-point hoists to provide single-point capability.

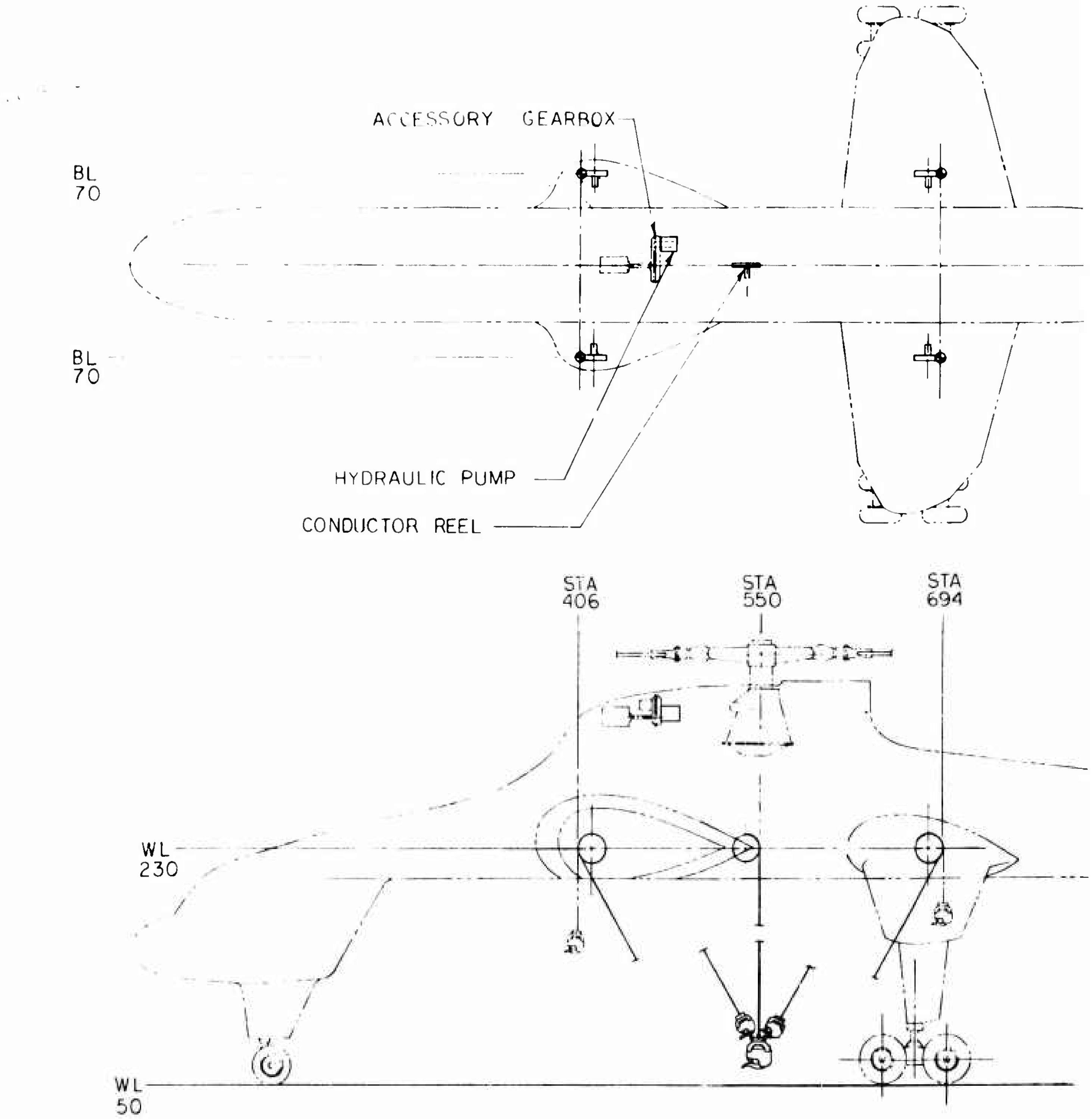
System Components and Weights

Four-Point Hoists (Type L, 4 required)	2200 Pounds
Hydraulic Pump	35
Hydraulic Motor	88
Plumbing and Oil	188
Hook and Swivel (40,000-lb Capacity)	192
Conductor Reel	50
Total System Weight	2753 Pounds
Single-Point Mission Weight	2753 Pounds
Multi-Point Mission Weight	2753 Pounds

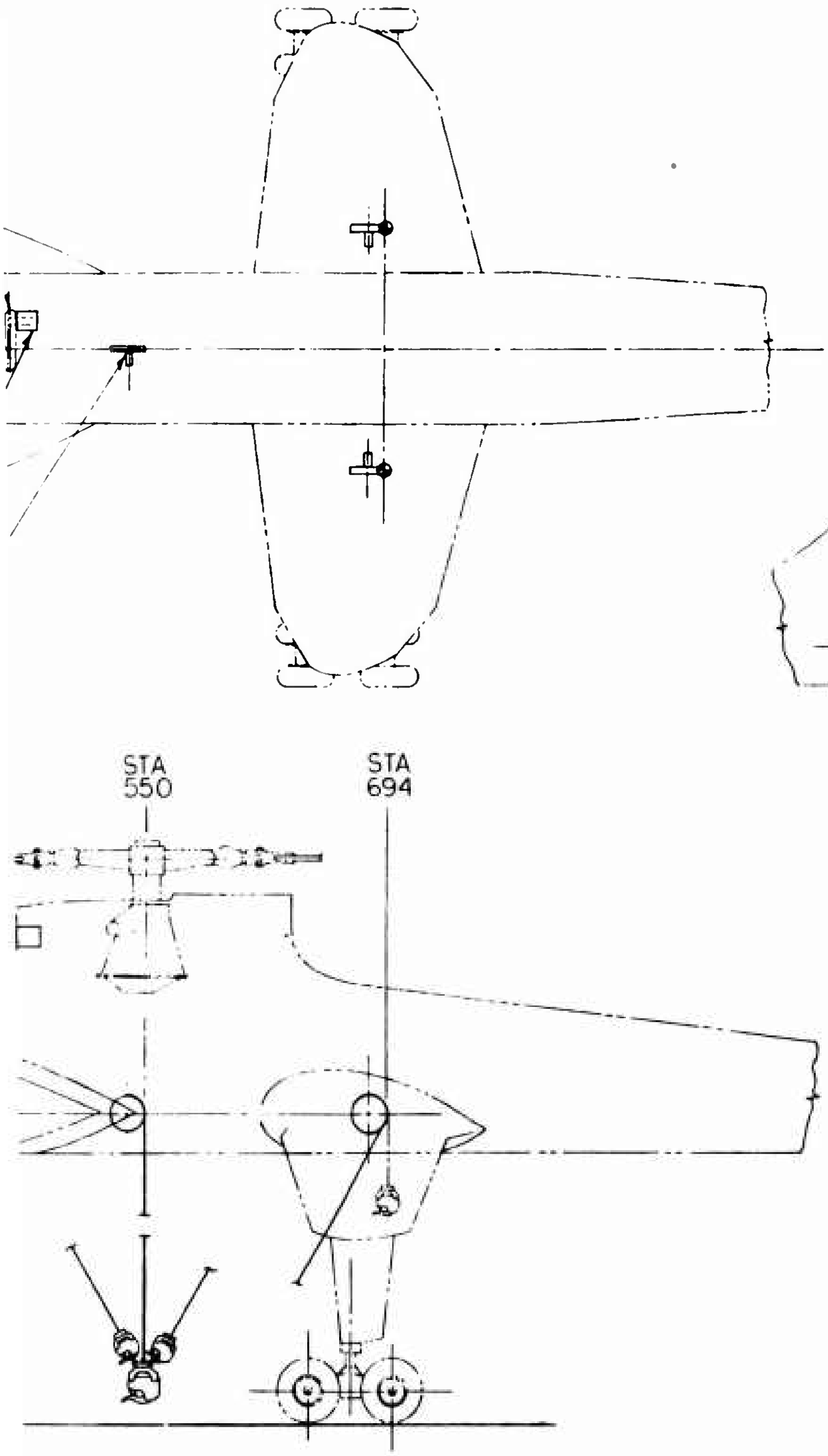
Power Required

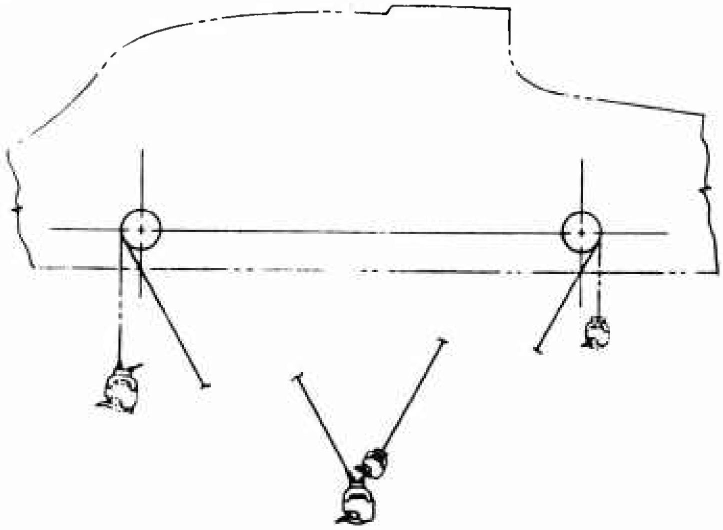
Single-Point Mission	-
Multi-Point Mission	171.8 HP

Figure 23. -7 Configuration.

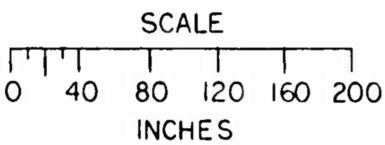

ACCESSORY GEARBOX
BL
70
BL
70
HYDRAULIC PUMP
CONDUCTOR REEL
STA
406
STA
550
STA
694
WL
230
WL
50

A.





-7A Configuration.



B.

-11 CONFIGURATION

General Description

Single-point mechanically driven. Hydraulically powered zero-moment hoists for the two-point system.

System Components and Weights

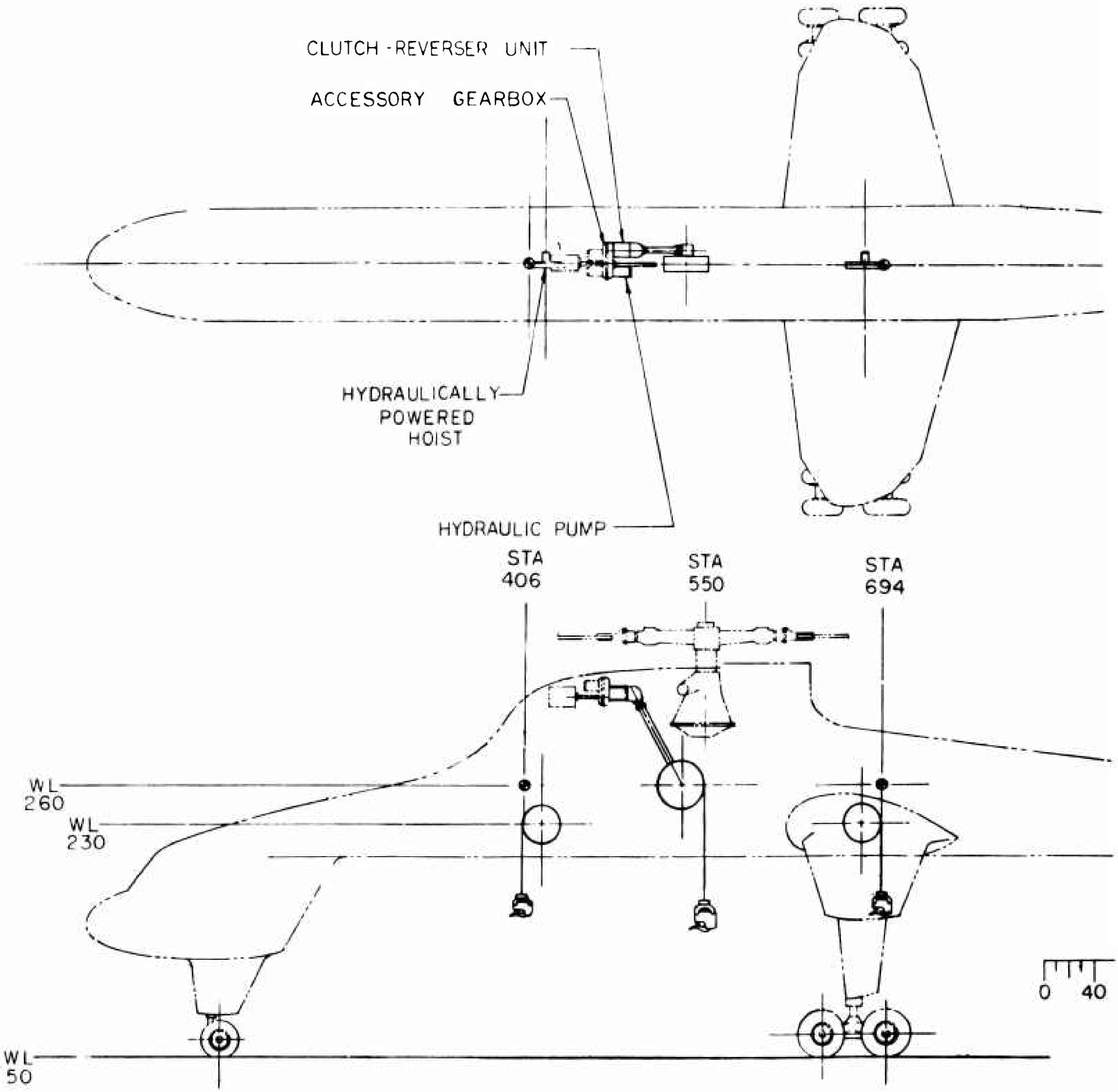
Single-Point Hoist (Type A)	1960 Pounds
Clutch-Reverser Unit	124
Two-Point Hoist (Type G, 2 required)	2004
Hydraulic Motors (2 required)	44
Angle Gearboxes (2 required)	43
Hydraulic Pump	35
Shafts and Couplings	6
Plumbing and Oil	174
Structure	40
Total System Weight	4430 Pounds
Single-Point Mission Weight (Remove two-point hoists)	2426 Pounds
Multi-Point Mission Weight (Remove single-point hoists)	2470 Pounds

Power Required

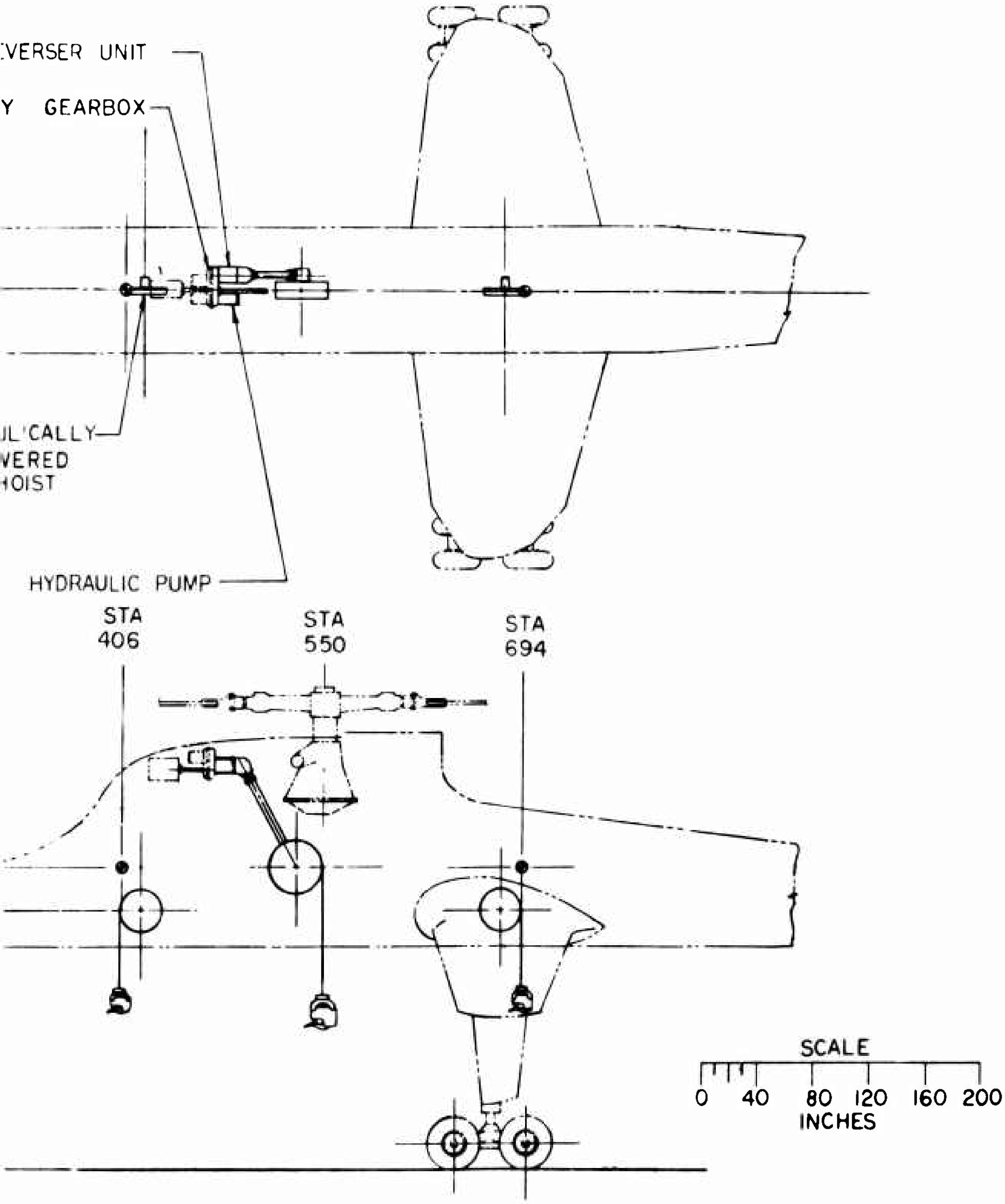
Single-Point Mission	94.0 HP
Multi-Point Mission	86.3 HP

Figure 24. -11 Configuration.

A.


CLUTCH-REVERSER UNIT
ACCESSORY GEARBOX
HYDRAULICALLY POWERED HOIST
HYDRAULIC PUMP
STA 406
STA 550
STA 694
WL 260
WL 230
WL 50
0 40

B.



VERSER UNIT
Y GEARBOX
IL'CALLY
VERED
HOIST
HYDRAULIC PUMP
STA
406
STA
550
STA
694
SCALE
0 40 80 120 160 200
INCHES

-13 CONFIGURATION

General Description

Single- and two-point hoists mechanically driven.

System Components and Weights

Single-Point Hoist (Type A)	1960 Pounds
Clutch-Reverser Unit	124
Two-Point Hoists (Type F, 2 required)	2044
Angle Gearboxes (4 required)	83
Shaft and Couplings	45
Clutches (3 required)	48
Structure	40
Total System Weight	4344 Pounds
Single-Point Mission Weight (Remove two-point hoists)	2300 Pounds
Multi-Point Mission Weight (Remove single-point hoist)	2384 Pounds

Power Required

Single-Point Mission	94.0 HP
Multi-Point Mission	61.1 HP

Figure 25. -13 Configuration.

-13 CONFIGURATION

General Description

Single- and two-point hoists mechanically driven.

System Components and Weights

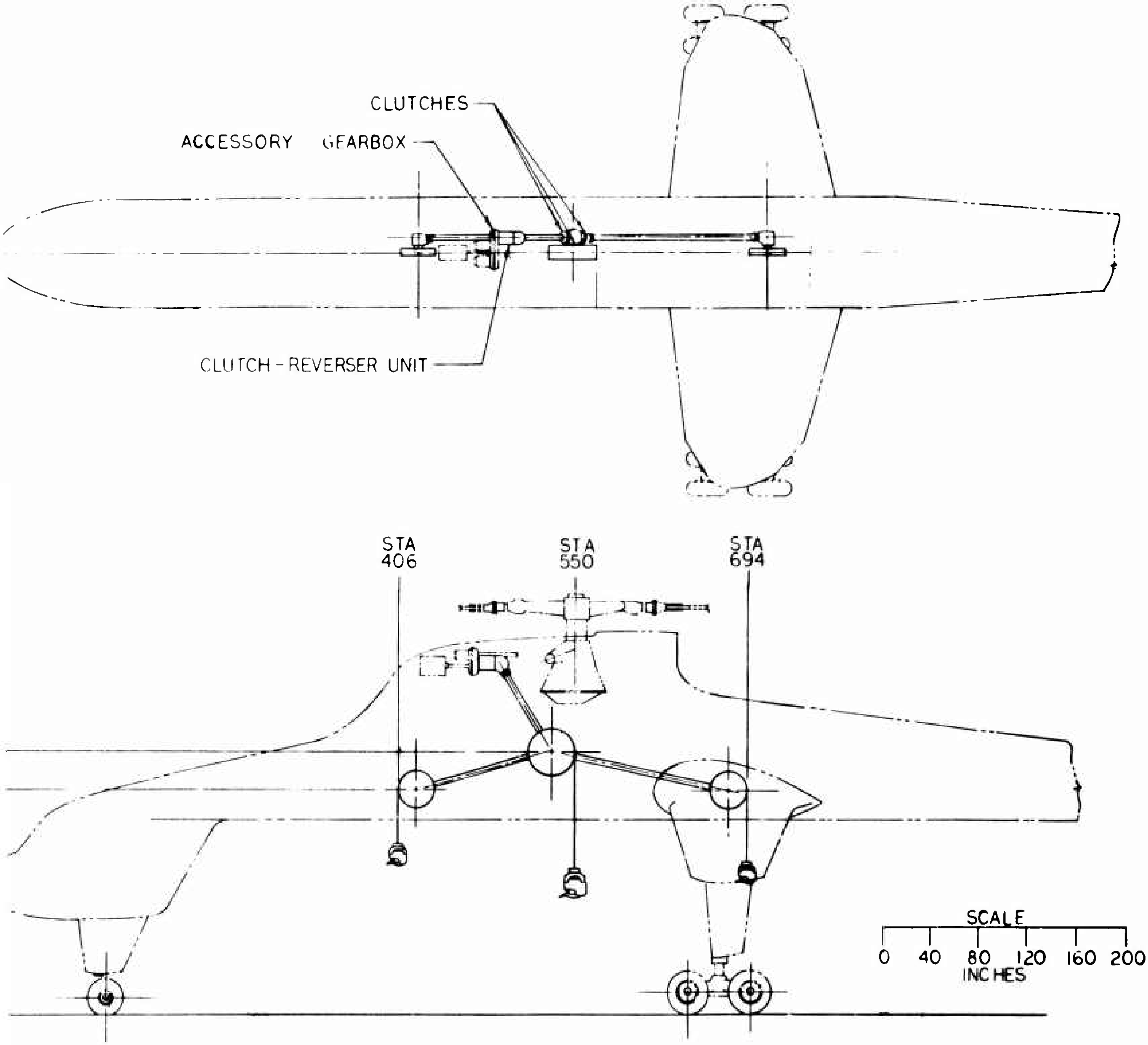
Single-Point Hoist (Type A)	1960 Pounds
Clutch-Reverser Unit	124
Two-Point Hoists (Type F, 2 required)	2044
Angle Gearboxes (4 required)	83
Shaft and Couplings	45
Clutches (3 required)	48
Structure	40
Total System Weight	4344 Pounds
Single-Point Mission Weight (Remove two-point hoists)	2300 Pounds
Multi-Point Mission Weight (Remove single-point hoist)	2384 Pounds

Power Required

Single-Point Mission	94.0 HP
Multi-Point Mission	61.1 HP

Figure 25. -13 Configuration.

A.



CLUTCHES
ACCESSORY GEARBOX
CLUTCH-REVERSER UNIT
STA 406
STA 550
STA 694
SCALE
0
40
80
120
160
200
INCHES

B.

-14 CONFIGURATION

General Description

Single-point hoist is a dual drum type mechanically driven. Cables are reeved over hydraulically powered traction sheaves for the two-point system.

System Components and Weights

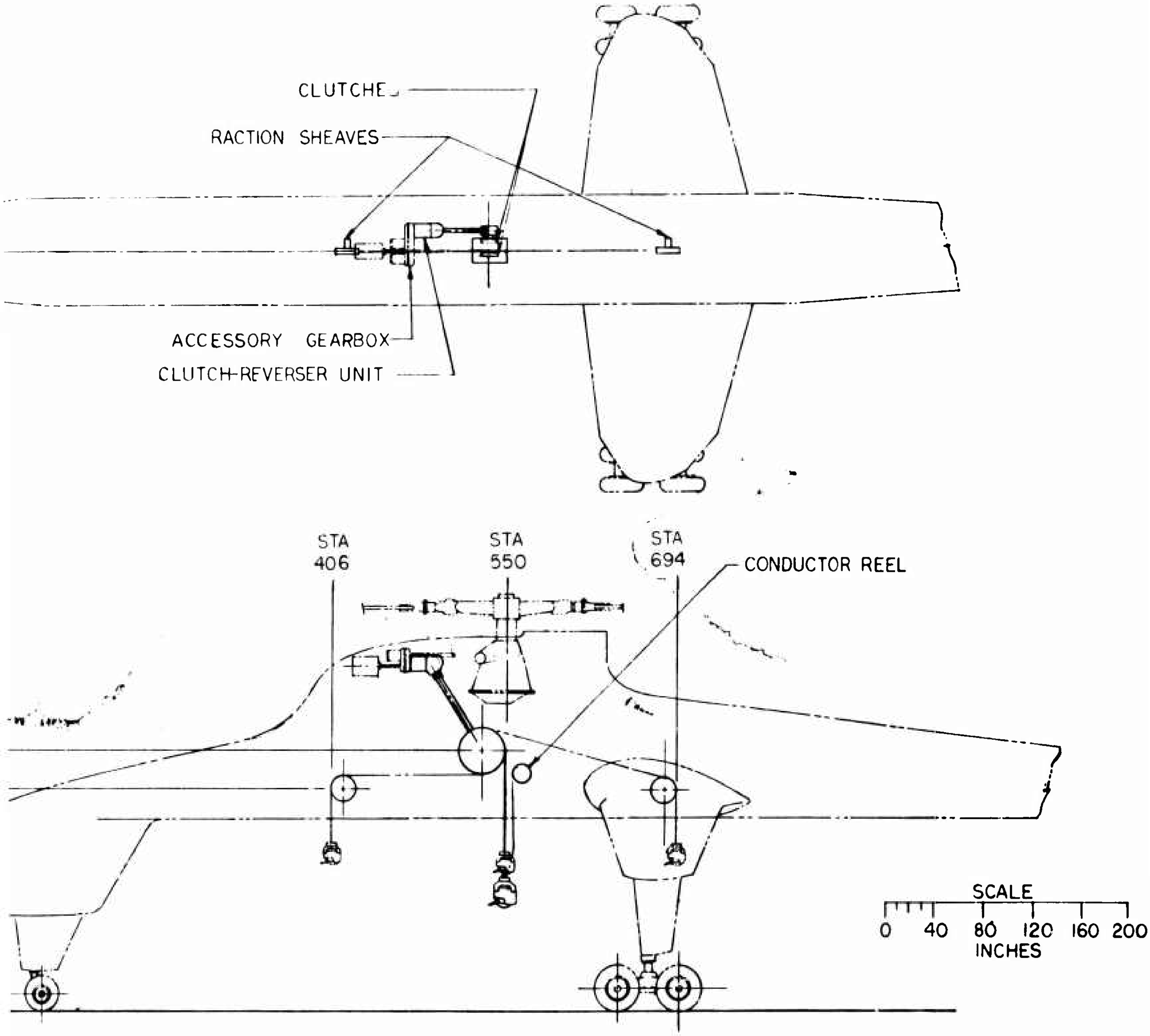
Dual Drum Single-Point Hoist (Type D)	2138 Pounds
Clutch-Reverser Unit	124
Hook and Swivel (40,000-lb Capacity)	192
Traction Sheaves (2 required)	120
Conductor Reel	50
Clutches (2 required)	32
Angle Gearboxes (2 required)	36
Shafts and Couplings	6
Plumbing and Oil	24
Structure	40
Total System Weight	2762 Pounds
Single-Point Mission Weight	2762 Pounds
Multi-Point Mission Weight (Remove 40,000-lb capacity hook)	2570 Pounds

Power Required

Single-Point Mission	90.6 HP
Multi-Point Mission	63.8 HP

Figure 26. -14 Configuration.

A


CLUTCHE_
RACTION SHEAVES
ACCESSORY GEARBOX
CLUTCH-REVERSER UNIT
STA 406
STA 550
STA 694
CONDUCTOR REEL
SCALE
0 40 80 120 160 200
INCHES

B.

-15 CONFIGURATION

General Description

No single-point hoist. Hydraulically powered zero-moment hoists for the two-point system with a beam, lockable to the aircraft, to provide single-point capability.

System Components and Weights

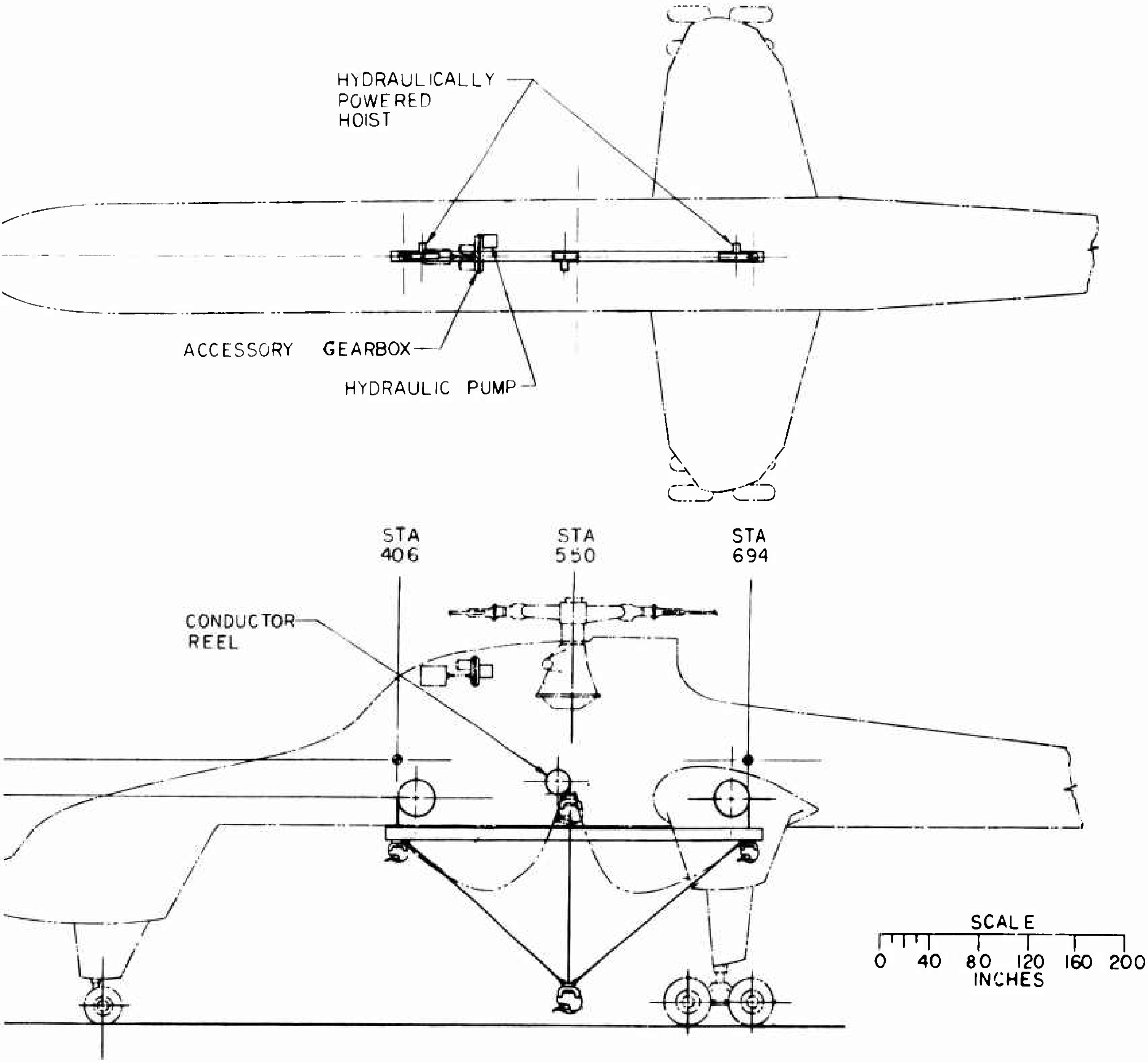
Two-Point Hooks (Type H, 2 required)	2144 Pounds
Suspension Beam with Hook & Slings)	1106
Conductor Reel	50
Hydraulic Pump	55
Hydraulic Motors (2 required)	44
Plumbing and Oil	174
Structure	40
Total System Weight	3613 Pounds
Single-Point Mission Weight	3613 Pounds
Multi-Point Mission Weight (Remove beam with hook & slings)	2507 Pounds

Power Required

Single-Point Mission	-
Multi-Point Mission	174.8 HP

Figure 27. -15 Configuration.

A.


HYDRAULICALLY POWERED HOIST
ACCESSORY GEARBOX
HYDRAULIC PUMP
STA 406
STA 550
STA 694
CONDUCTOR REEL
SCALE
0 40 80 120 160 200
INCHES

B.

-17 CONFIGURATION

General Description

No single-point hoist. Mechanically powered conventional hoists with cable joined to a common hook to provide single-point capability.

System Components and Weights

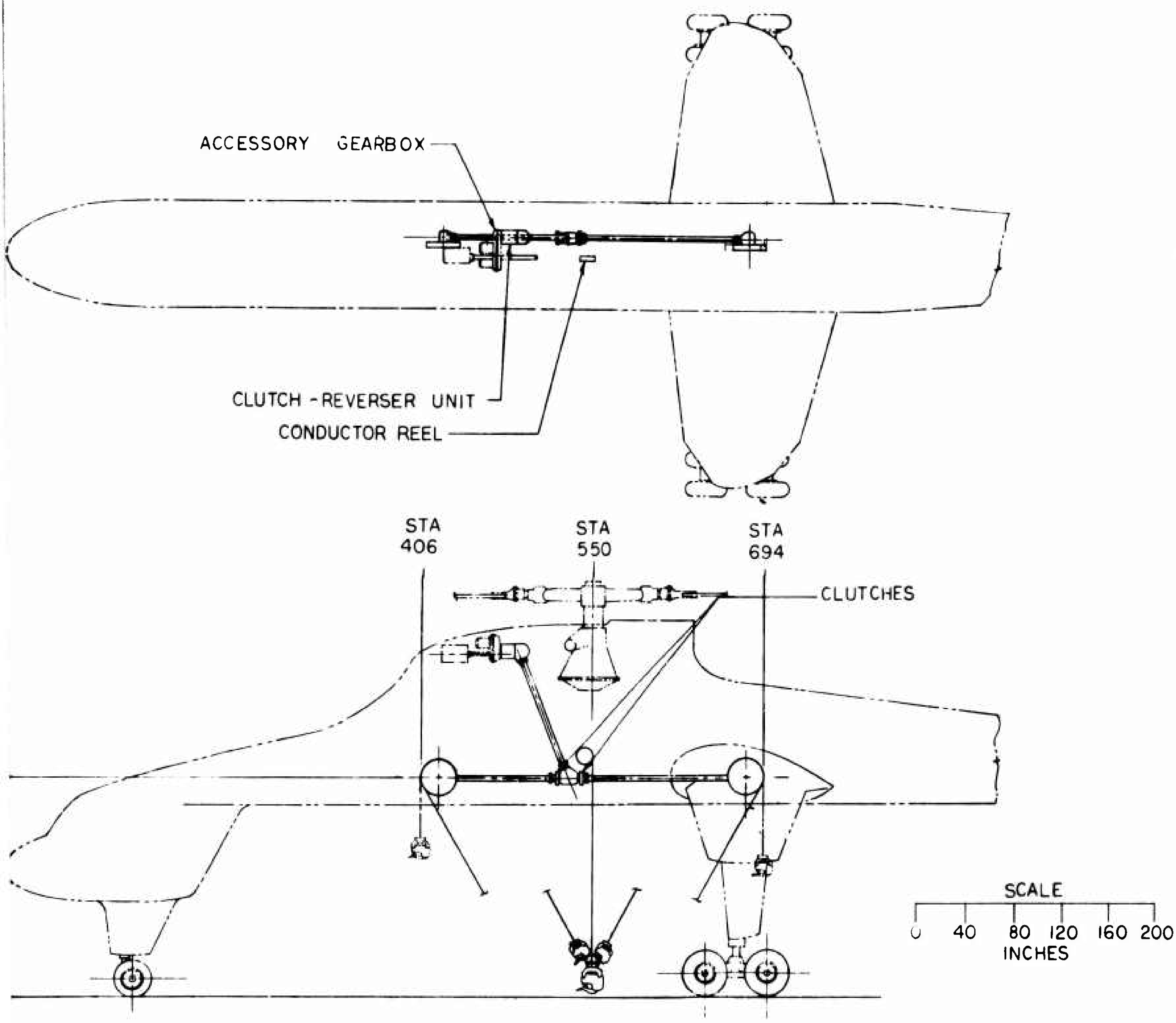
Two-Point Hoists (Type F, 2 required)	2144 Pounds
Clutch-Reverser Unit	124
Angle Gearboxes (4 required)	60
Shafts and Couplings	38
Clutches (2 required)	32
Conductor Reel	50
Hook and Swivel (40,000-lb Capacity)	192
Structure	40
Total System Weight	2680 Pounds
Single-Point Mission Weight	2680 Pounds
Multi-Point Mission Weight (Remove 40,000-lb capacity hook)	2488 Pounds

Power Required

Single-Point Mission	
Multi-Point Mission	123.9 HP

Figure 28. -17 Configuration.

A.


ACCESSORY GEARBOX
CLUTCH - REVERSER UNIT
CONDUCTOR REEL
STA 406
STA 550
STA 694
CLUTCHES
SCALE
0 40 80 120 160 200
INCHES

B.

-18 CONFIGURATION

General Description

No single-point hoist. Mechanically powered conventional hoists with cables reeved over hydraulically powered traction sheaves and joined to master hook for single-point capability.

System Components and Weights

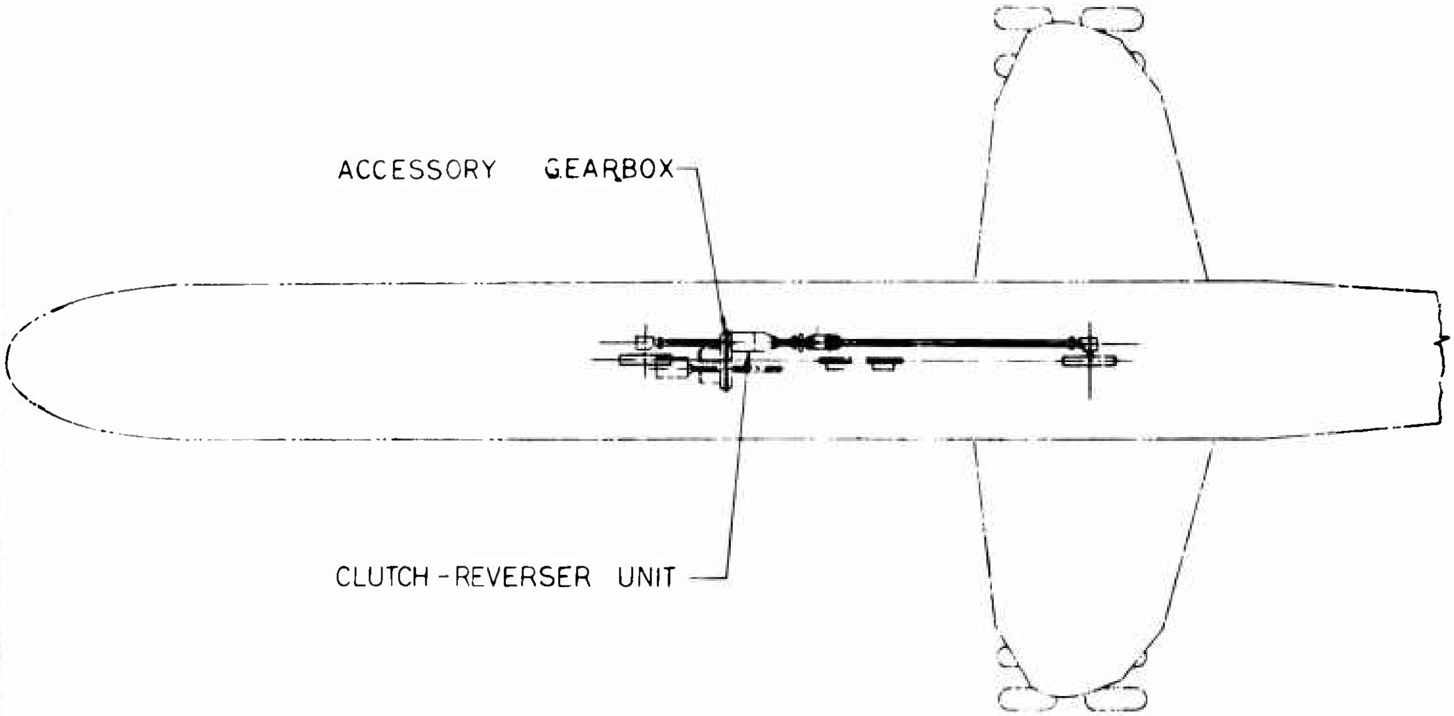
Two-Point Hoists (Type F, 2 required)	2144 Pounds
Clutch-Reverser Unit	124
Traction Sheaves (2 required)	120
Hook and Swivel (40,000-lb Capacity)	192
Angle Gearboxes (4 required)	50
Shafts and Couplings	40
Clutches (2 required)	16
Plumbing and Oil	6
Structure	40
Total System Weight	2732 Pounds
Single-Point Mission Weight	2732 Pounds
Multi-Point Mission Weight	2732 Pounds

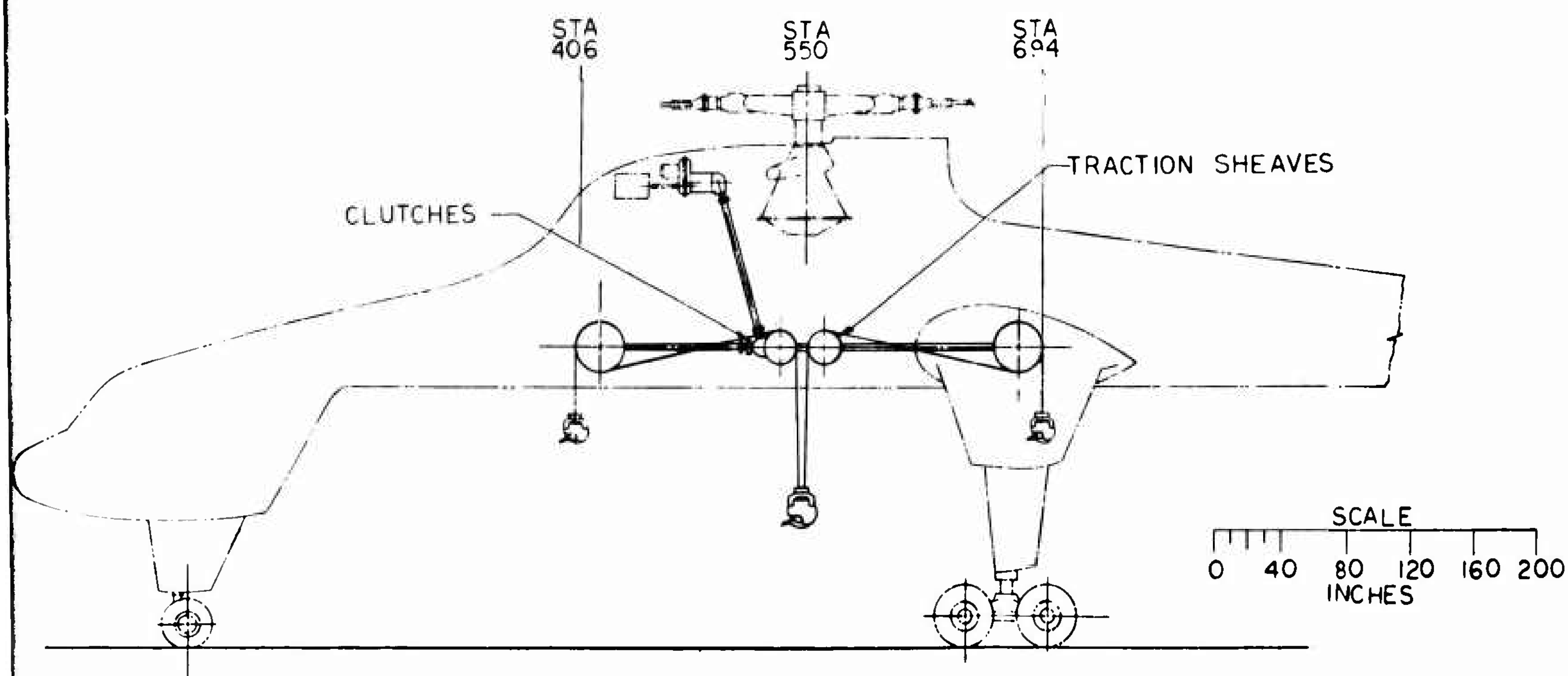
Powered Required

Single-Point Mission	-
Multi-Point Mission	125.1 HP

Figure 29 -18 Configuration.

A.


ACCESSORY GEARBOX
CLUTCH - REVERSER UNIT


STA 406
STA 550
STA 694
TRACTION SHEAVES
CLUTCHES
SCALE
0 40 80 120 160 200
INCHES

B.

TABLE XVII

SUMMARY
SINGLE-POINT PLUS FOUR-POINT
LOAD SUSPENSION CONFIGURATIONS

Config. Number	Description	Total System Weight* (lb)	Drive	Hoi Typ
-1	Single-point hoist mechanically driven. Hydraulically powered zero moment hoists for the four-point system.	4396	Mech.	Con
-2	Single-point hoist hydraulically powered by one or two motors. Hydraulically powered zero-moment hoists for the four point system.	4287	Hyd.	Con
-3	Single- and four-point hoists mechanically driven.	4213	Mech.	Con
-4	Single-point hoist mechanically driven. Two mechanically driven, dual drum hoists with cables reeved over hydraulically powered traction sheaves for the four-point system.	4430	Mech.	Con
-5	No single-point hoist. Hydraulically powered zero-moment hoists for the four-point system with a frame, lockable to the aircraft, to provide single-point capability	3733	-	-
-6	No single-point hoist. Hydraulically powered zero-moment hoists with cables joined to a common hook for single point capability and reeved over hydraulically powered traction sheaves for the four-point system.	2991	-	-
-7	No single-point hoist. Hydraulically powered zero-moment hoists for the four-point system. Cables joined by a master hook carried by one of the four-point hoists to provide single-point capability.	2753	-	-

*Add 103 pounds for single-point hoist with
Add 362 pounds for single-point hoist with

A.

FOUR-POINT
FIGURATIONS

Single Point					Four Point				
Drive	Hoist Type	Cable Length (ft)	HP Reqd	System Weight* (lb)	Drive	Hoist Type	Cable Length (ft)	Total HP Reqd	System Weight (lb)
Mech.	Conv.	80	94.2	2396	Hyd.	Zero Mom.	50	84.8	2436
Hyd.	Conv.	80	132.5	2287	Hyd.	Zero Mom.	50	84.8	2327
Mech.	Conv.	80	94.2	2261	Mech.	Conv.	50	58.0	2253
Mech.	Conv.	80	94.1	2450	Mech.	Dual Drum & Traction Sheaves	50	64.9	2470
-	-	-	-	3733	Hyd.	Zero Mom.	80	171.8	2561
-	-	-	-	2991	Hyd.	Zero Mom.	80	179.1	2751
-	-	-	-	2753	Hyd.	Zero Mom.	80	171.8	2753

hoist with 100 feet of cable
hoist with 150 feet of cable

B.

TABLE XVIII

SUMMARY
SINGLE-POINT PLUS TWO-POINT
LOAD SUSPENSION CONFIGURATIONS

Config. Number		Total System Weight* (lb)	Drive	Hoist Type
-11	Single-point mechanically driven. Hydraulically powered zero-moment hoists for the two-point system.	4430	Mech.	Conv.
-13	Single- and two-point hoists mechanically driven.	4344	Mech.	Conv.
-14	Single-point hoist a dual drum type mechanically driven. Cables reeved over hydraulically powered traction sheaves for the two-point system.	2762	Mech.	Dual Drum
-15	No single-point hoist. Hydraulically powered zero-moment hoists for the two-point system with a beam, lockable to the aircraft, to provide single-point capability.	3613	-	-
-17	No single-point hoist. Mechanically powered conventional hoists with cables joined to a common hook to provide single-point capability.	2680	-	-
-18	No single-point hoist. Mechanically powered conventional hoists with cables reeved over hydraulically powered traction sheaves and joined to a master hook for single-point capability.	2732	Hyd.	Traction Sheav

*Add 103 pounds for single-point hoist witl
Add 362 pounds for single-point hoist witl

A.

II

TWO-POINT
FIGURATIONS

Single Point					Two Point				
Drive	Hoist Type	Cable Length (ft)	HP Reqd	System Weight* (lb)	Drive	Hoist Type	Cable Length (ft)	Total HP Reqd	System Weight (lb)
Mech.	Conv.	80	94.0	2426	Hyd.	Zero Mom.	50	86.3	2470
Mech.	Conv.	80	94.0	2300	Mech.	Conv.	50	61.1	2384
Mech.	Dual Drum	80	90.6	2762	Hyd.	Traction Sheaves	-	63.8	2570
-	-	-	-	3613	Hyd.	Zero Mom.	80	174.8	2507
-	-	-	-	2680	Mech.	Conv.	88	123.9	2488
Hyd.	Traction Sheaves	-	-	2732	Mech.	Conv.	92	125.1	2732

nt hoist with 100 feet of cable
nt hoist with 150 feet of cable

B.

LOAD ACQUISITION AND RELEASE

SINGLE-POINT MODE

The single-point mode offers the most versatile and safest method for acquiring and releasing external loads. It is equally adaptable to ground and hovering type pickups. Since the cargo hook is connected to the cable by means of a swivel assembly, there is little resistance to rotation of loads being lifted. Because of this feature, bulky loads will have to be carried below the main landing gear; less bulky loads may be snugged up as close as in any of the multi-point systems to permit higher forward speeds. The effect of oscillating loads on the stability of the aircraft is minimized, since the hoist is located close to the center of gravity of the aircraft. This is more fully discussed in the AIRCRAFT - LOAD INTERACTION section. Hovering pickup, by any of the methods requiring the use of a beam to convert the multi-point system to single point (as in the -5 and -15 configurations), will present inherent hazards to both ground crew and to vehicles during both pickup and release. Therefore, the use of these systems is not recommended. The physical size of the hook, which weighs 192 pounds, will require that the vehicle sling be carried to the hook. A short leader line from the sling will facilitate this type of hookup and enable the ground crewman to hook up without having to climb to the top of the vehicle to make the connection.

In-flight release of cargo will be possible by using the electrical hook release. Use of tandem-dual cable cutters (see page 54) will ensure that cargo can be jettisoned in the event of a malfunction of the electrical release. Normal load release will be made after the cargo is put on the ground. Four methods of load release are possible: two for the pilot and two for the ground crew. The pilot can open the hook either by the electrical release, or, in the event of emergency, by shearing the cable with the tandem-dual cable cutters. The ground crewman can open the hook with the manual release knob or can slide the load ring off the hook by retracting the keeper on the hook in the event of a malfunction of the manual load release. An automatic touchdown release, whereby the hook automatically opens when the cargo is put on the ground, can be incorporated. This feature adds another release mode and provides greater redundancy.

Towing by the single-point hoist requires the use of special equipment of the type used on the CH-54A (see Figure 30, page 92) if cable loads are expected to exceed 18,000 pounds. This limitation is detailed in the AIRCRAFT-LOAD INTERACTION section, page 96.

TWO-POINT MODE

The two-point mode is adaptable to either ground or hovering pickups, depending primarily upon the type of terrain for the method to be used. Ground pickup offers the safest method and should be used whenever circumstances permit.

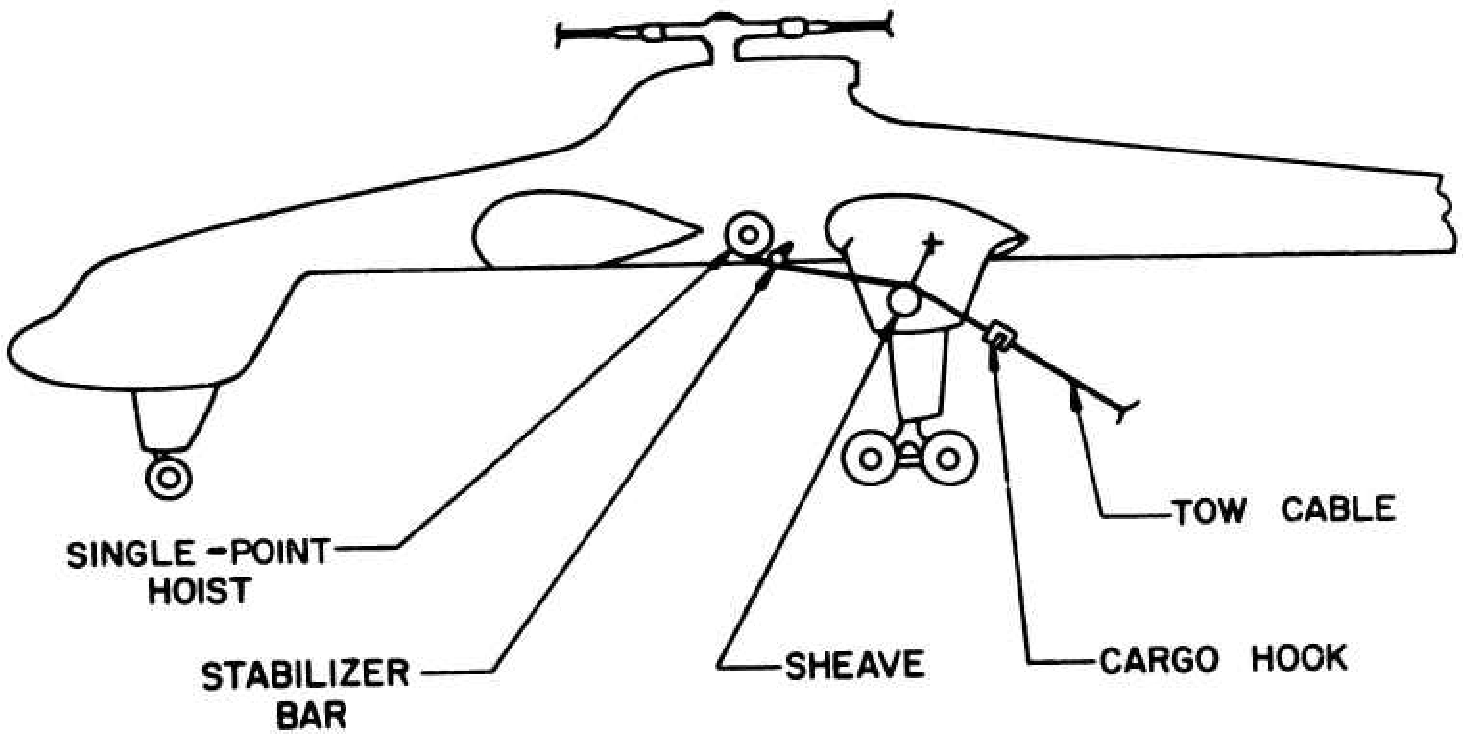


Figure 30. Special Tow Gear - H.L.H.

Two-point hovering pickups are inherently more risky than those made in the single-point mode. This is because of the possibility that upsetting loads could be transmitted to the aircraft if it should drift and cause one cable to become tight before the other was attached. This risk can be largely eliminated by reeling out enough cable to put the hooks on the ground with adequate slack to permit attachment to the cargo. Since the hooks weigh 87 pounds, leader lines from the cargo slings, or vehicle attachment points, will greatly facilitate hookup. These lines can be attached to the hooks instead of requiring the hooks to be carried to the attachment points. Lifting cargo with a C.G. located midway between pick-up points from a hovering attitude will be performed with the load sensitive control energized. When cable loads are equalized, the synchronized lift control will be engaged and the load can be snugged into position. Beeping (individual control of hoists) is available to allow control of cargo that has an asymmetrical center of gravity. These control systems have been fully described in the HOIST SYSTEM AND COMPONENTS DESIGN section page 12.

In-flight trimming of loads to compensate for the effects of the aerodynamic drag can be attained by engaging the load sensitive control. The advisability of making such in-flight adjustment at other than hovering or at very low forward speed conditions is questionable. Preliminary analysis indicates that multi-point loads will assume a stable aerodynamic position for reasonably adjusted cable lengths at any given forward speed. In-flight changes in cable lengths may affect aircraft stability and tend to produce ptiching oscillations. Further evaluation utilizing wind tunnel tests is desirable to obtain qualitative data upon which the limitations or advisability can be based.

Normal load release will consist of synchronized lowering of the load to the ground from either a hover or the landed position of the aircraft. After the cables are slackened, the hooks can be opened electrically by the pilot or manually by the ground crew. In the unlikely event of both electrical and manual release failure, a ground crewman can slide the load ring off the hook load beam by retracting the spring loaded keeper on the hook. In the event that no ground crewman is on-site, it will be necessary to keep a slight tension (a cable tension indicator is provided) on the cables to permit the load to pull off the hook in the electrical release mode. This is necessary since the hook load beam is spring loaded for automatic relatching. As in the single-point mode, a tandem- dual cable cutter on each hoist will provide emergency release by shearing both cables.

The addition of a cockpit controlled, manual hook release, if practical, will not permit safe in-flight jettisoning of loads by hook release. Only by the combination of electrical and manual hook release motion in the cockpit could this be considered as a possible method to be used. Even this combination of motions should not be considered as an acceptable means of in-flight jettisoning, since malfunction of the hook unlocking mechanism could still occur. Only by use of the tandem-dual cable cutters mounted on the hoists can a safe in-flight load jettison be performed.

Towing is feasible within the 24,000-pound capability of the aft hoist. It is not feasible to use both hoists for greater tow capabilities because of the difficulties of obtaining equal cable loads.

FOUR-POINT MODE

The four-point system is equally adaptable to ground or to hovering pickup. Ground pickup offers the safest method and should be used whenever circumstances permit. As in the two-point mode, it should be standard procedure during hovering pickups to have the hooks on the ground with adequate slack in the cables to permit their attachment to cargo. Since the hooks weigh 59 pounds, it is possible to attach the hooks to vehicle pickup points without the use of leader cables which are required for both the single- and two-point systems. Use of leader cables may be desirable, however.

Lifting cargo with a C.G. located midway between pickup points from hovering attitude would be performed with the load sensitive control energized. When cable loads are equalized, the synchronized lift control will be engaged and the load will be snugged into position. Beeping (individual control of hoists) is available to allow control of cargo that has an asymmetrical center of gravity. Lifting cargo with a C.G. that is not symmetrically located with respect to the pickup points could result in the cargo assuming an extreme angle with the ground (see Figure 12, page 46).

In-flight trimming of loads to compensate for the effects of aerodynamic drag of the cargo can be attained by engaging the load sensitive control. The advisability of making such an adjustment, as more fully discussed on page 111, is questionable. Normal load release will consist of synchronized lowering of the load to the ground from either a hover or the landed position of the aircraft. After cable slack is observed, the hooks can be opened electrically by the pilot or manually by the ground crew. As in the two-point system, a ground crewman can slide the load ring off the hook load beam by retracting the spring loaded keeper on the hook in the event of failure of the other release systems. If no ground crewmen are available, it is necessary to keep some tension (a cable tension indicator is provided) on the cables to permit the load to pull off the hook when released from the cockpit. As in the single- and two-point systems, tandem-dual cable cutters provide emergency release.

To achieve maximum safety, it is recommended that in-flight load release be accomplished by shearing the cables. It is not considered safe with present stage of the art release systems to attempt an electrical hook release in flight because of the inherent risk if one or more hooks fail to open.

Should one or more of the hooks fail to open, the entire load would be transferred to the cables supporting these hooks and could cause the aircraft to become uncontrollable. Since many loads to be carried may be well within the ultimate (failing) strength of one or more of the cables, it is not possible to assume cable failure as a backup release system.

In-flight release of any multi-point load should therefore be accomplished only by use of the tandem-dual cable cutters.

Limited towing (12,000 pound maximum) can be accomplished by using the left rear hoist. Any method of towing requiring that two or more hoist cables be joined presents inherent load sharing problems.

AIRCRAFT - LOAD INTERACTION

STABILITY OF SLUNG LOADS

The stability of the load is the major limitation on forward speed when carrying slung loads. An oscillating or spinning load can transmit periodic forces to the aircraft which are detrimental to performance and handling qualities. High density spherical or cube shaped loads, such as a cargo net filled with ammunition, are generally aerodynamically stable and do not impose limitations on the aircraft. Low density, nonsymmetrical loads, such as a helicopter fuselage, are aerodynamically unstable and require some type of stabilizing device. Stabilization can be accomplished by multi-point suspension or by use of a small parachute attached to the load through a swivel joint.

The major advantage of a multi-point suspension system is the restoring moment it generates when the load is displaced in yaw. With either the two- or four-point suspension system, pods can be pulled snug against the aircraft, thus completely eliminating the yaw divergence problems. This stability contribution of multi-point systems deteriorates as cable length increases. Figure 31, page 97, shows the change in static directional restoring stability, N_ψ, with cable length for both the two- and four-point systems with a typical load of 25,000 pounds.

$$N_\psi = W \frac{(x^2 + y^2)}{57.3L} \text{ ft-lb per degree} \qquad (23)$$

where

W is the load, pounds

x is the longitudinal distance between cable attachment points, feet

y is the lateral distance between cable attachment points, feet

L is the vertical distance between the load and cable attachment point, feet

At yaw divergence speed, the restoring torque of the system just equals the unstable aerodynamic moment of the load. Figure 32, page 98, shows the variation of static directional stability of a helicopter fuselage with forward speed when slung at various distances below the fuselage. It can be seen that the four-point system provides a benefit of 5 knots in forward speed over that of a two-point system. Beyond 60 knots, a stabilizing device would be mandatory.

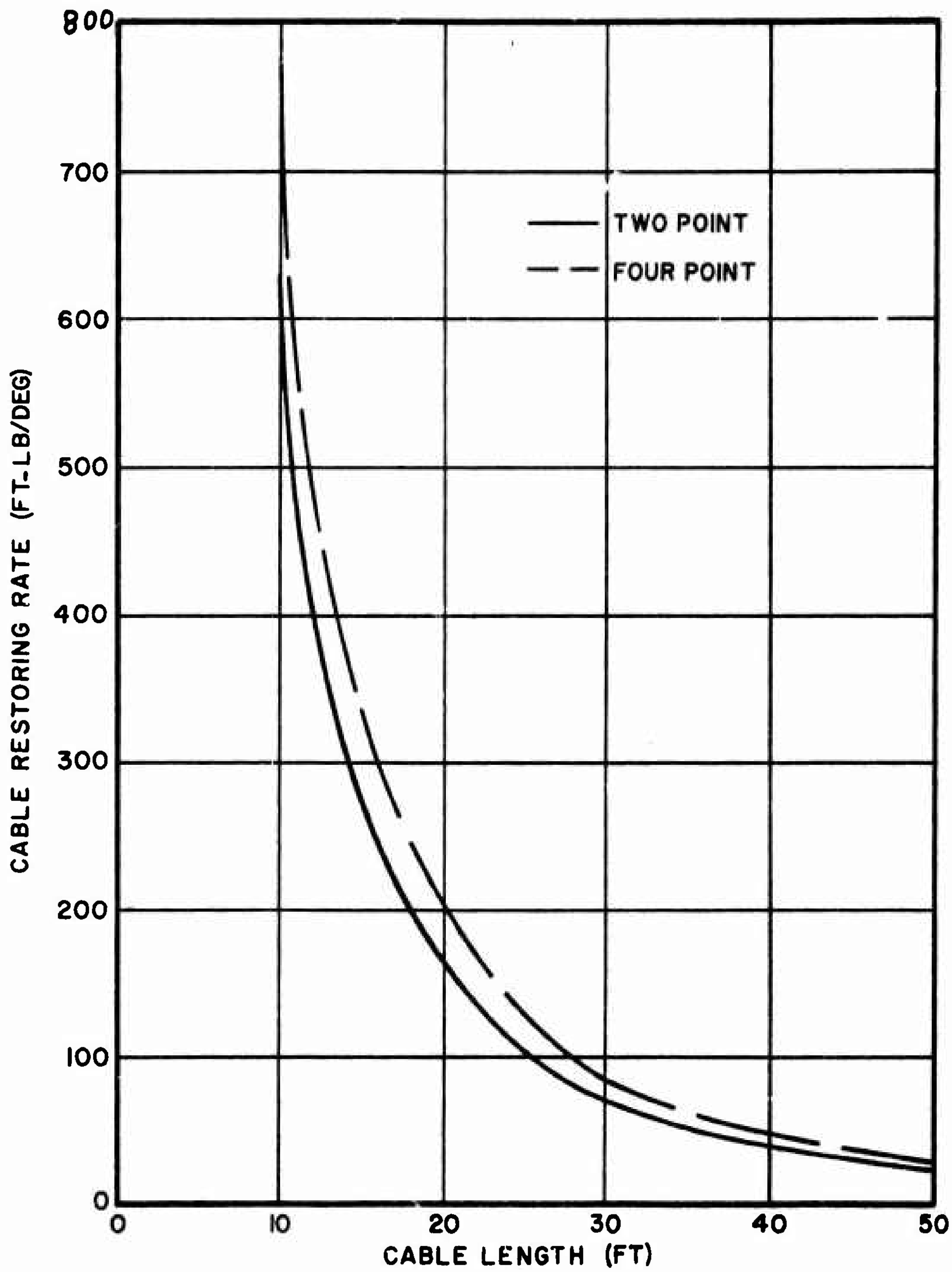


Figure 31. Restoring Moment for Two- and Four-Point Suspension with 25,000-Pound Load.

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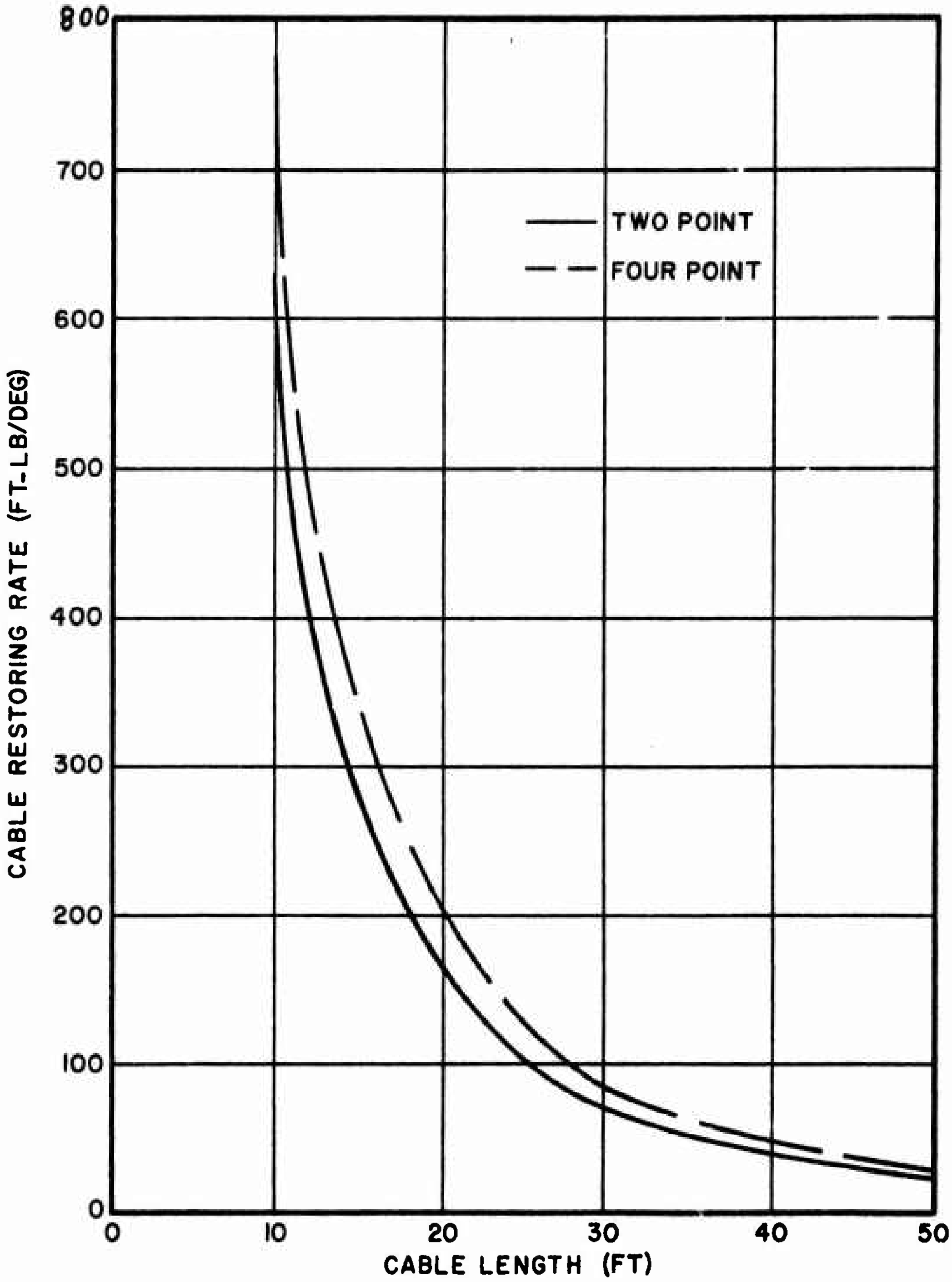


Figure 31. Restoring Moment for Two- and Four-Point Suspension with 25,000-Pound Load.

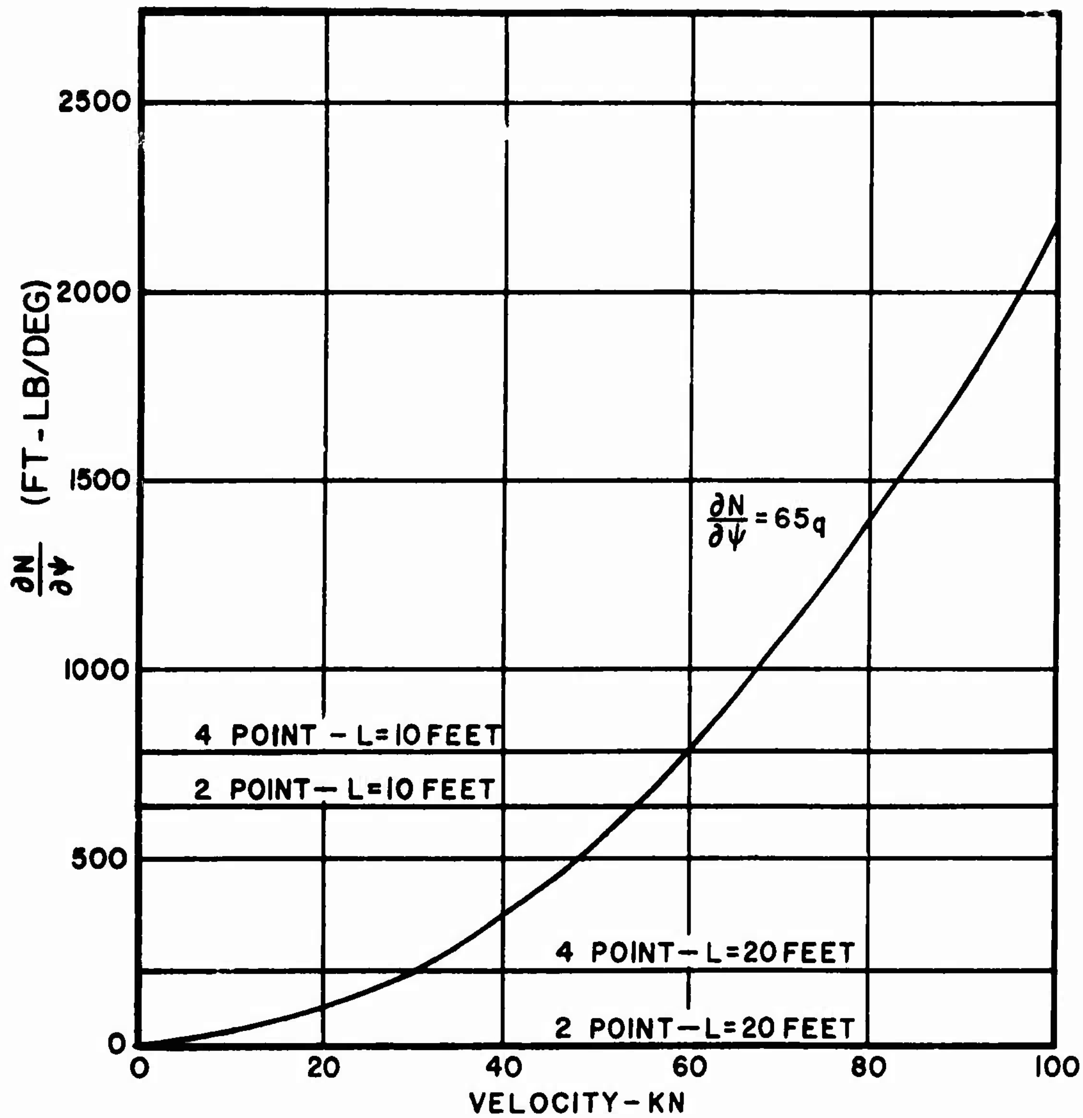


Figure 32. Typical Yaw Divergence of 25,000-Pound Helicopter Fuselage.

The loads considered as typical are listed in Table XIX, page 101. Although actual wind tunnel data are not available for these vehicles, it is felt that high density loads, such as the personnel carrier and the self-propelled mortar, will need some additional stabilization in the single-point mode and none in either of the multi-point modes. Based on wind tunnel tests of stores made for the S-60 Flying Crane, the 5-ton wrecker will require a drag parachute for stabilization at speeds of 100 knots on all suspension systems. The 155 mm howitzer will require added stabilization in the single-point system and little, or none, on either of the multi-point systems.

CENTER-OF-GRAVITY SHIFT

None of the loads evaluated will present any C.G. problems for either the single- or tandem-rotor aircraft. For the single-rotor type, all C.G.'s are within the F.S. 526 to 574 allowable limits. With the exception of the 5-ton wrecker on the multi-point systems, all C.G.'s are at, or near, F.S. 550. In this case the overall C.G. is at F. S. 569 within allowable limits. Similarly, for the tandem-rotor type, all C.G.'s are within the F.S. 527 to 589 allowable limits. Except for the 5-ton wrecker on the multi-point systems, all C.G.'s are at F.S. 557. In this case the overall C.G. is at F.S. 576, which is within allowable limits.

TOWING CAPABILITY

Towing characteristics of the single-rotor aircraft were calculated with the aid of a computer program. Figure 33, page 100, shows trim control settings for a zero skew angle (aircraft plane of symmetry parallel to and coincident with the direction of the tow force). Experience in towing with the RH-3A, and in particular pilot's comments, dictates maximum pitch and roll attitudes of -22° and + 10° respectively. Figure 33 indicates that there will be no difficulty in meeting the roll attitude requirement but that the pitch attitude restriction limits the maximum tow cable tension to 18,000 pounds. In this situation the cable angle, relative to the earth, is 6°, tow cable length is 475 feet, and altitude is 50 feet. Lateral and longitudinal control positions show adequate control margin; trim positions are such that the pilot can execute a recovery in case the tow cable is suddenly released.

An increase in tow capability would be possible if special towing gear (see Figure 30, page 92) were fitted to the aircraft. This gear will move the tow cable reaction point further aft and is similar to that used on the CH-54A.

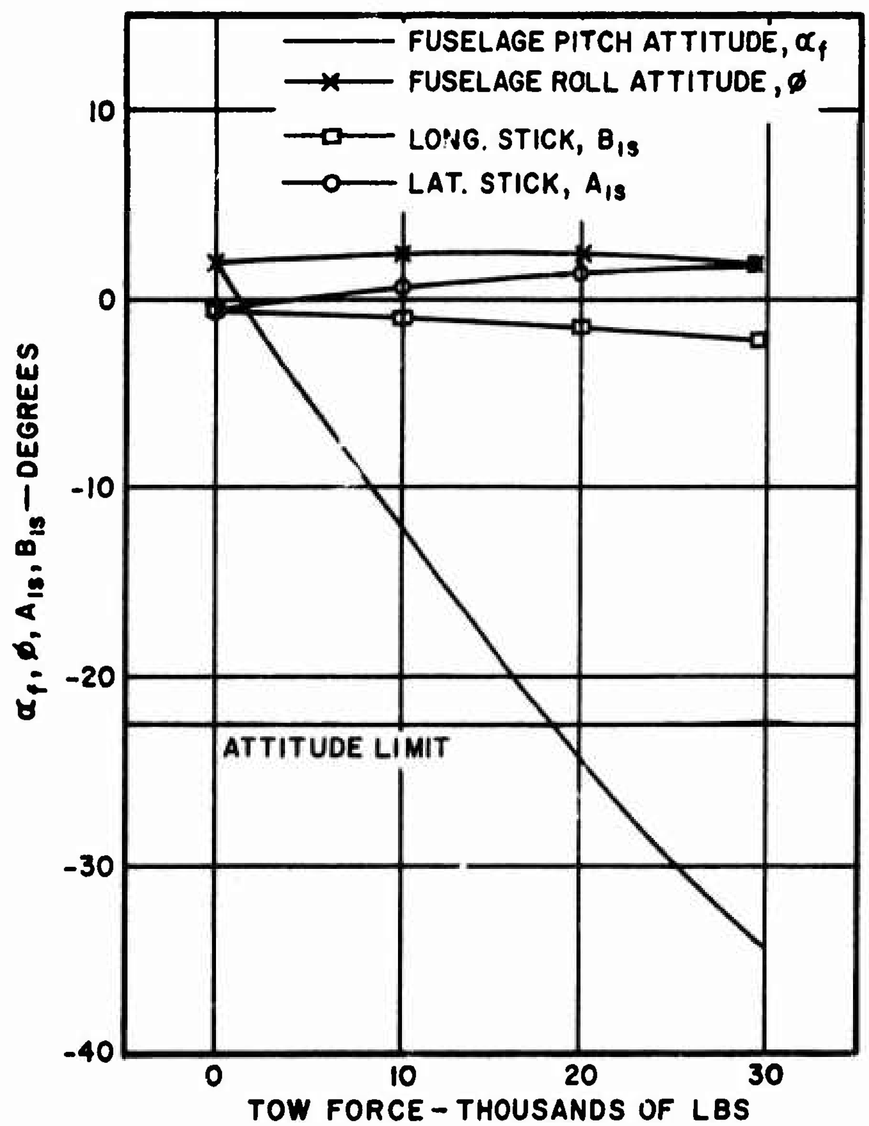


Figure 33. Low-Speed Towing Characteristics (Gross Weight 38,000 Pounds, C.G. at Sta. 550, Zero Skew Angle).

AIRCRAFT CONTROLLABILITY

Neither the single- nor the tandem-rotor aircraft should experience any trim difficulties during load acquisition or normal load release with the single or multi-point systems. Figure 34, page 102, shows trim control positions for the single-rotor type both with and without external loads. A parasite drag correction was made to the computer program for each load. No pitching moment corrections were made, as the lines of action of moment contributing forces are at, or very near, the aircraft C.G. The parasite drag corrections are in reasonable agreement with wind tunnel tests; estimated parasite drag corrections are shown in Table XIX.

TABLE XIX
ESTIMATED PARASITE DRAG

Item No. *	Vehicle	Parasite Drag(ft^2)
38	155 MM Howitzer	18
49	Personnel Carrier	16.7
83	5-Ton Wrecker	49
86	Self-Propelled Mortar	33

*Item number in Appendix I

As shown in Figure 34, significant changes from the basic aircraft trim are required for some loads. If the load should be jettisoned, the aircraft would respond as if a sudden input were applied to the controls. Figures 35 and 36, pages 103 and 104, are time history relationships to the aircraft equations of motion for the more adverse trim situations in Figure 34. The aircraft will rise rapidly after the self-propelled mortar is jettisoned from a hovering attitude and at 60 knots forward speed, as shown in Figures 35 and 36. In both situations, a reduction in collective pitch is necessary to prevent an excessive rate of climb.

Figures 37 and 38, pages 105 and 106, show that the most critical situations in pitch for jettisoning of the 5-ton wrecker are in hover and at 60 knots forward speed. As seen from these time histories, the pitch response is controllable with the automatic stabilization equipment engaged. Due to the basic instability of the aircraft in pitch, jettisoning of the load with the automatic stabilization equipment off should be accompanied by a corrective control input.

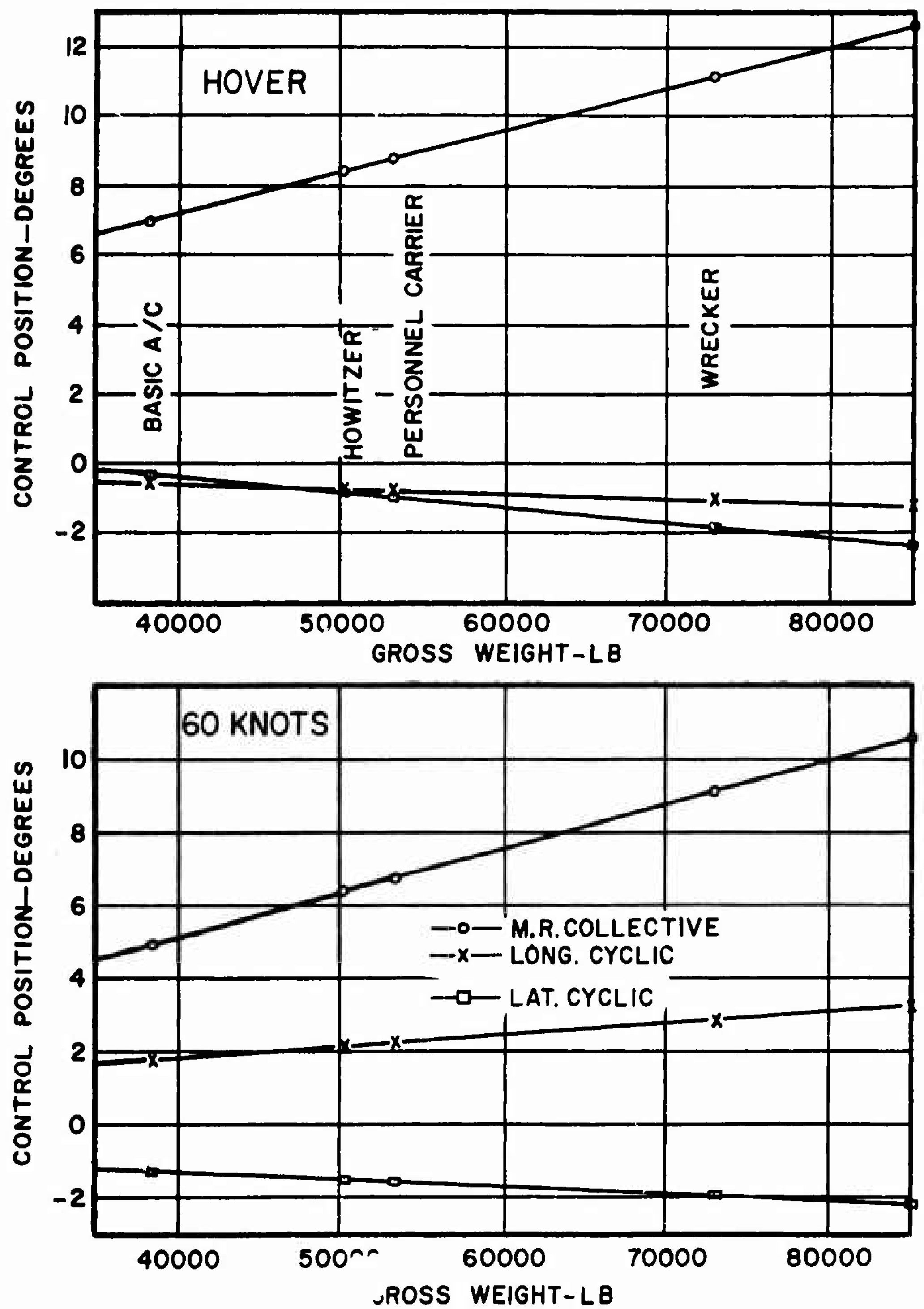


Figure 34. Trim Characteristics vs Gross Weight - Single Rotor.

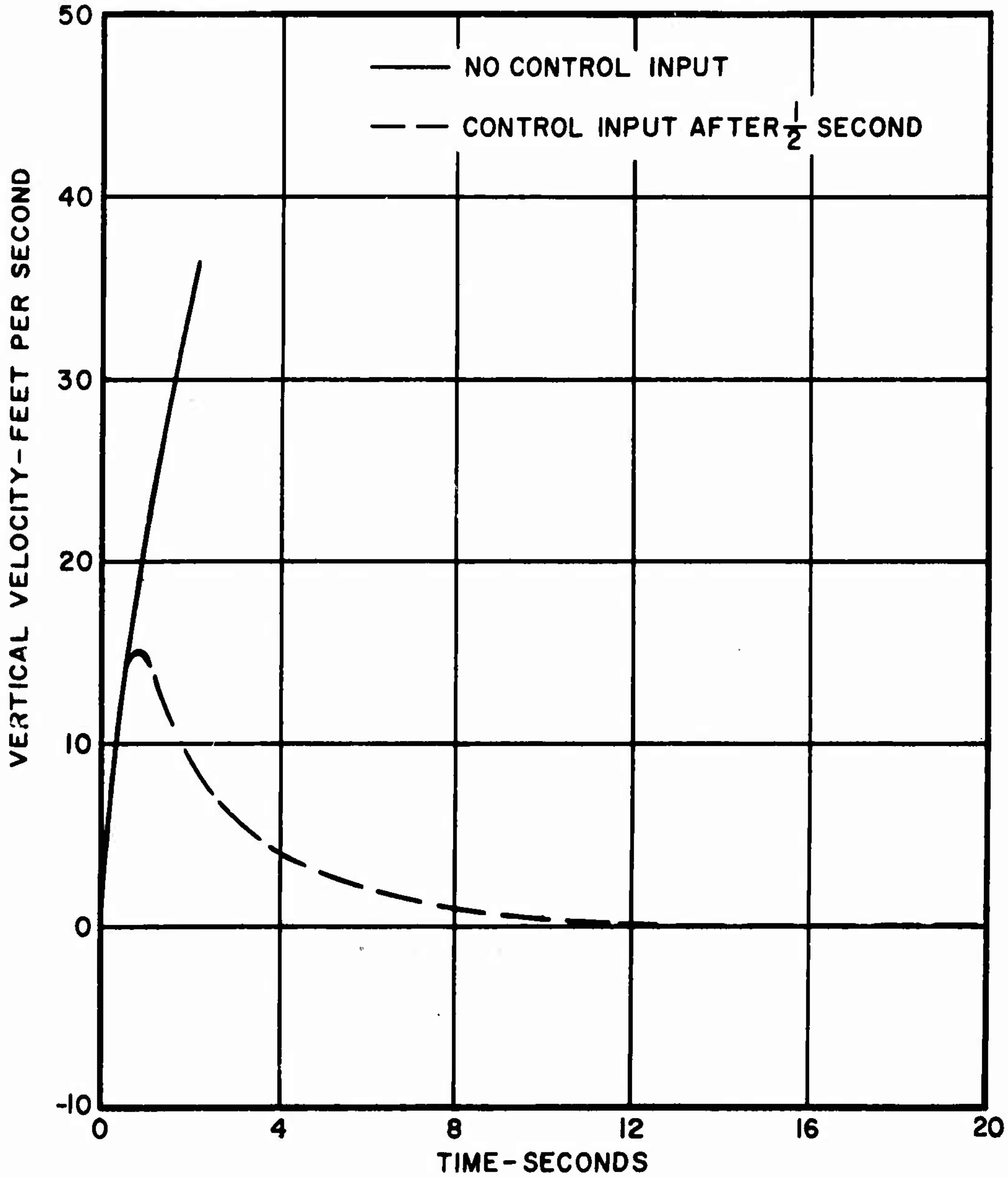


Figure 35. Vertical Response to Release of Self-Propelled Mortar in Hover.

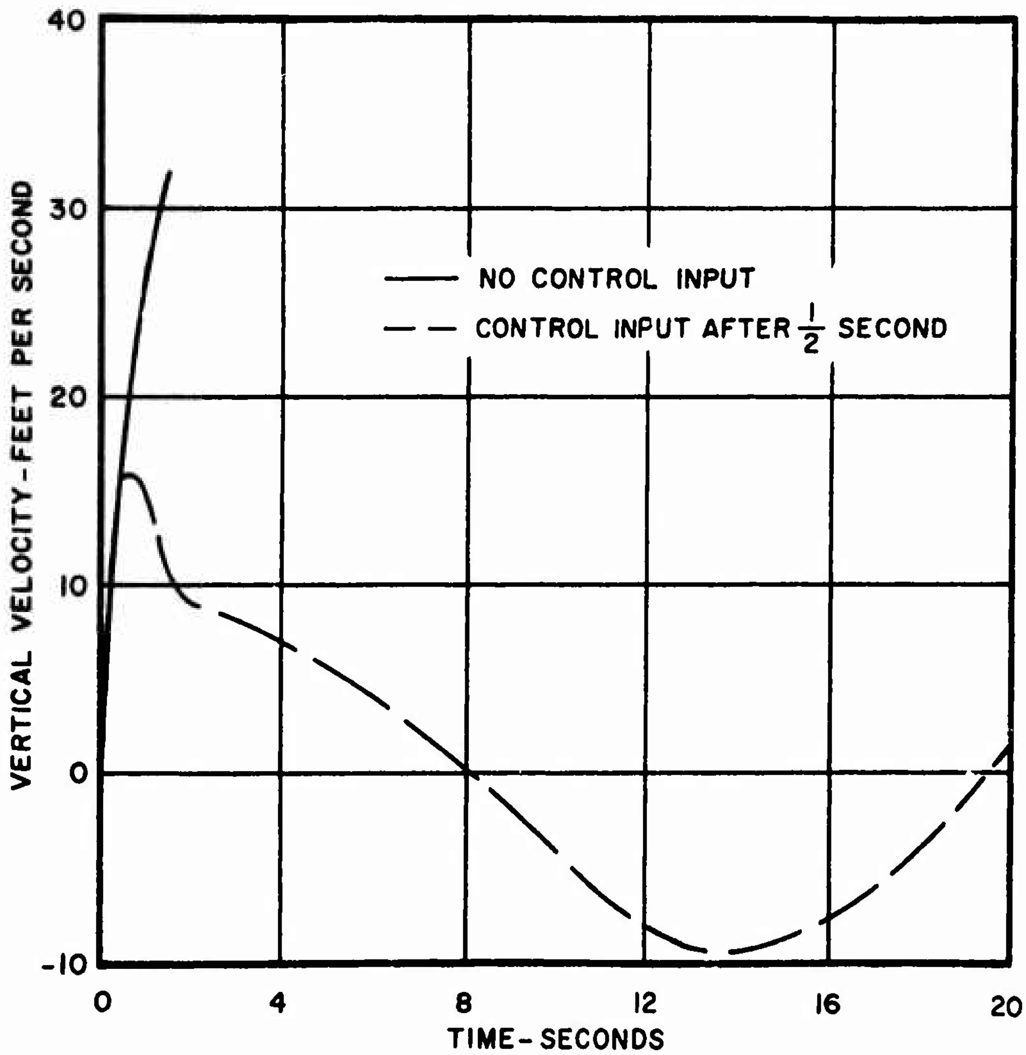


Figure 36. Vertical Response to Release of Self-Propelled Mortar at 60 Knots.

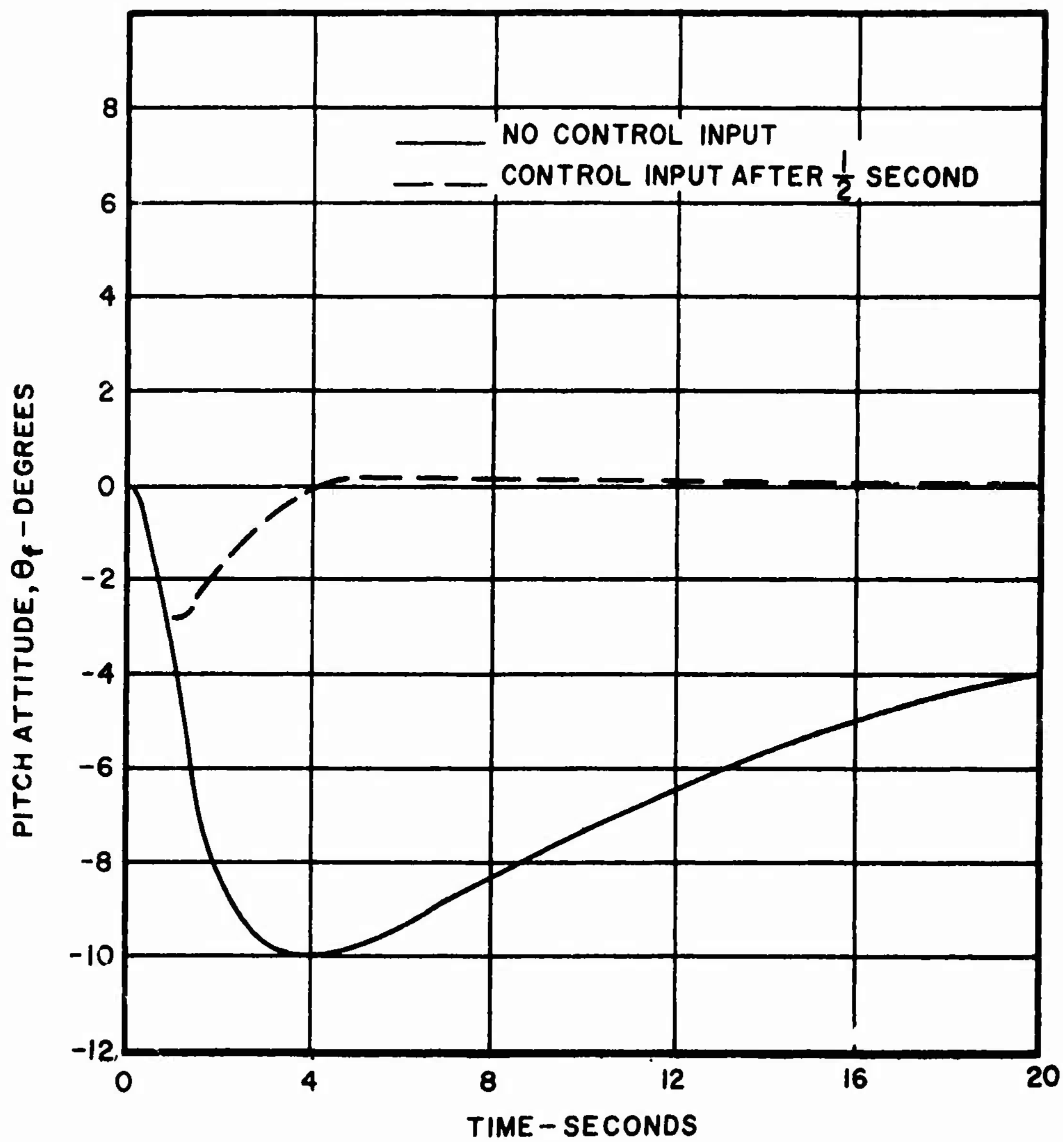


Figure 37. Pitch Response to Release of 5-Ton Wrecker in Hover.

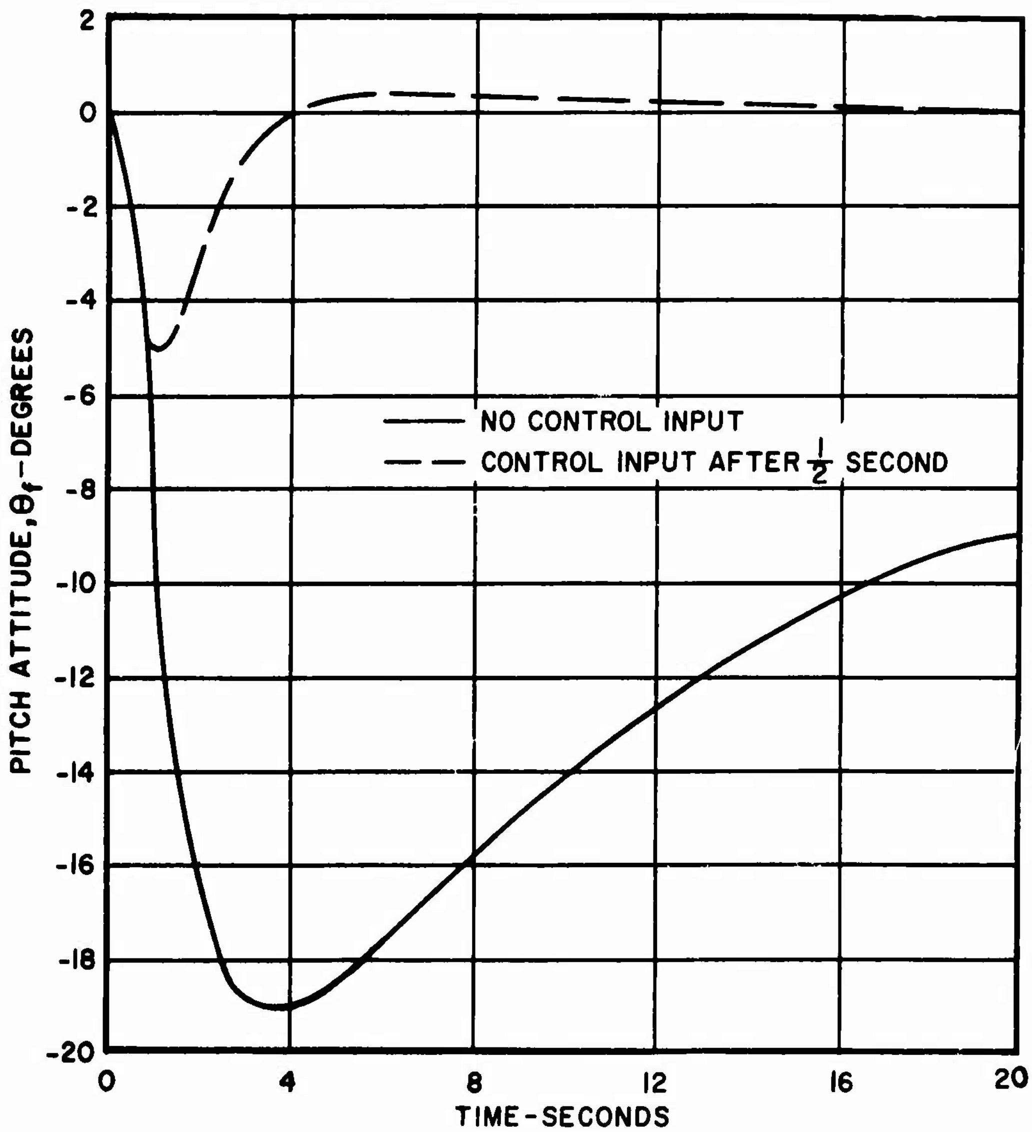


Figure 38. Pitch Response to Release of 5-Ton Wrecker at 60 Knots.

Neither single- nor tandem-rotor aircraft should exhibit any adverse trim changes when reeling in, or out, the entire length of cable on the single-point hoist if cable travel limitations are kept within proper limits. In order to keep resulting stick motion equal to or less than that corresponding to values on the CH-54A, it is necessary to require the use of a double layer drum hoist if a 150-foot cable length is required.

VERTICAL OSCILLATION

Divergent vertical oscillation (vertical bounce), potentially inherent in all heavy lift helicopters carrying external cargo, is dynamically a forced response of the aircraft's first fuselage mode and coupled cargo mode at one times main rotor (1p) excitation frequency. Although the aircraft's basic fuselage mode frequency may be well removed from its 1p operating speed, the attachment of a relatively large load though a flexible suspension cable can shift the fuselage bending mode to within the 1p range, as shown in Figure 39, page 108.

Depending on the ratio of the load mass to that of the fuselage, the input parameters, and the amount of inherent system damping, the resulting fuselage response can vary from small, convergent, uncomfortable cockpit levels to amplitudes divergent to aircraft structural integrity. Therefore, it is necessary to analyze the coupled load suspension/fuselage bending mode characteristics and to incorporate positive control to decouple the first fuselage and load suspension modes.

Dynamic decoupling is achieved in the heavy-lift helicopter by incorporating an isolator with variable stiffness characteristics as a function of load, as shown in Figure 40, page 109. The isolator provides essentially constant first fuselage and load suspension vertical frequencies, both well separated from 1p frequency excitation throughout the load application range (see Figure 41, page 110). These results were analyzed by using a Sikorsky Aircraft free vibration program on the IBM 7094 computer using 70 degrees of freedom, as shown in Figure 42, page 112. Vertical, longitudinal, and pitch motions are included in the programming analysis.

The heavy lift helicopter isolator requirements of Figure 40 are similar to those of the CH-54A. The isolator on the CH-54A has been flight-tested and proven to be effective in providing the required dynamic decoupling. Therefore, a similar isolator configuration is planned for the heavy lift helicopter.

POD JETTISON

In the event of an in-flight emergency, such as loss of one or more engines, it may be desirable to jettison a cargo loaded pod in order to increase the probability of effecting a safe emergency landing. Such a procedure is possible, when the pod is supported by the multi-point hoists, by shearing the cables. Preliminary analysis indicates that there should be no resulting pod pitching problems and that the pod should clear the tail cone

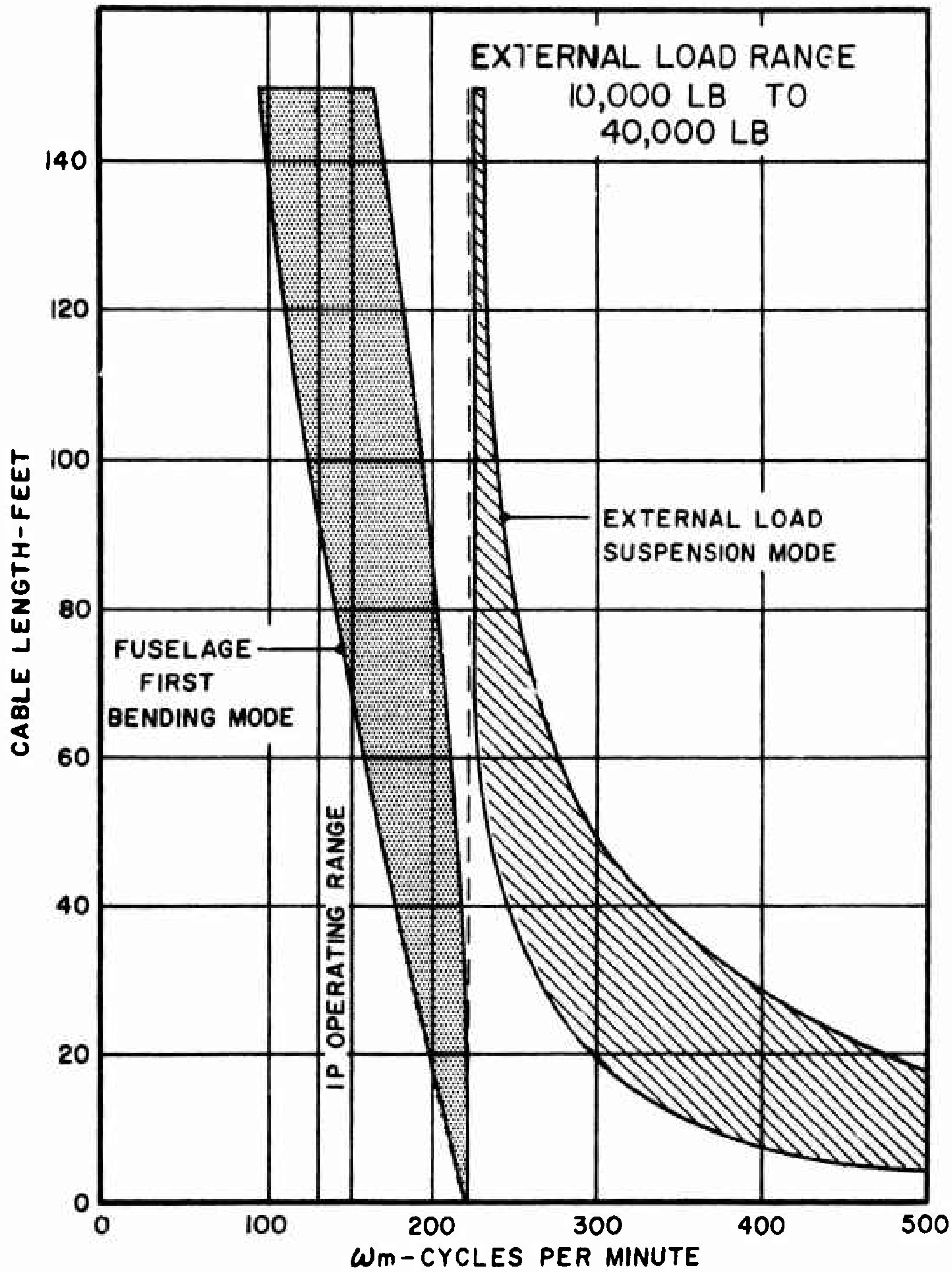


Figure 39. Vertical Bounce Mode Frequencies vs Cable Length Without Decoupler.

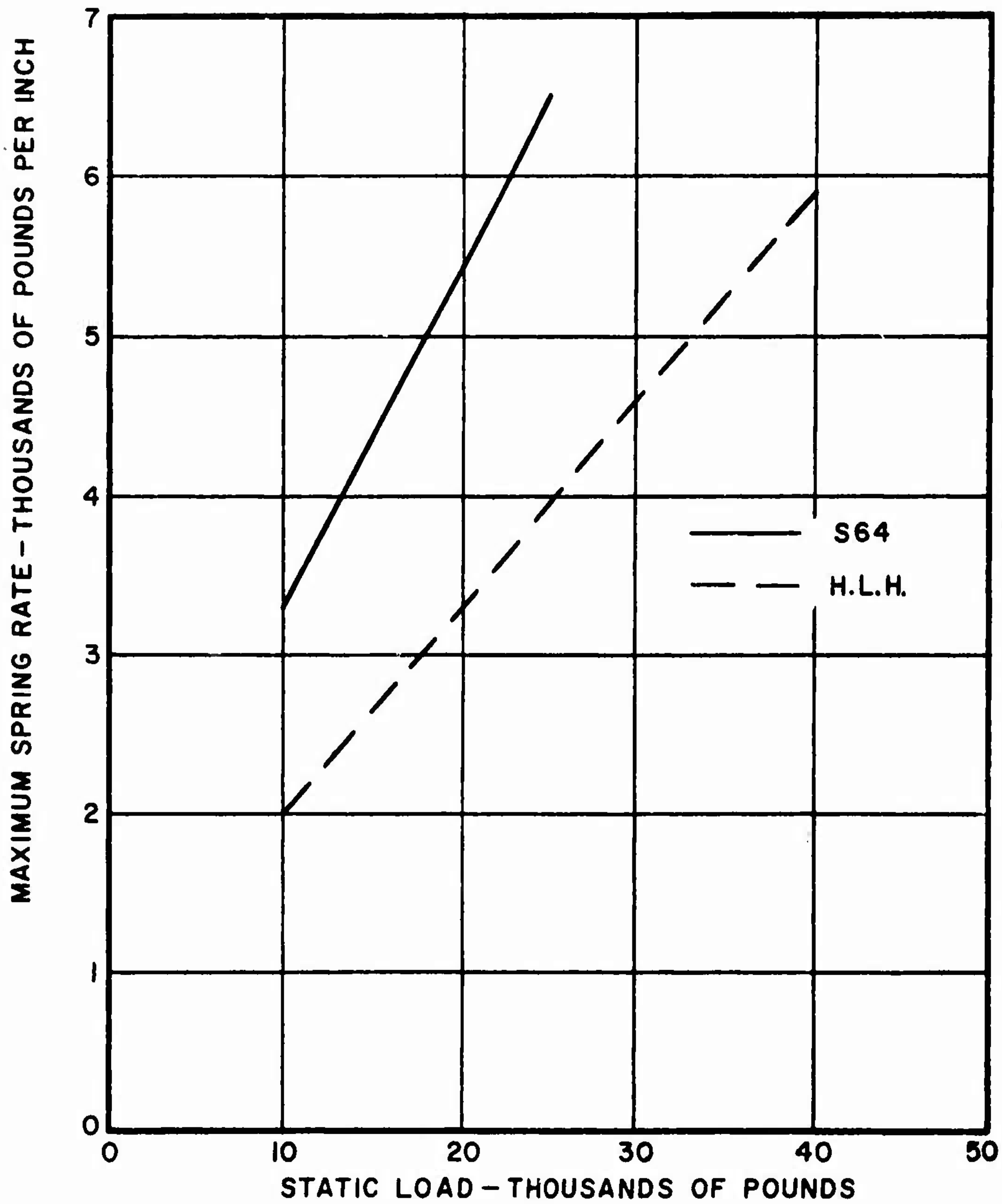


Figure 40. Decoupler Spring Rate vs Load.

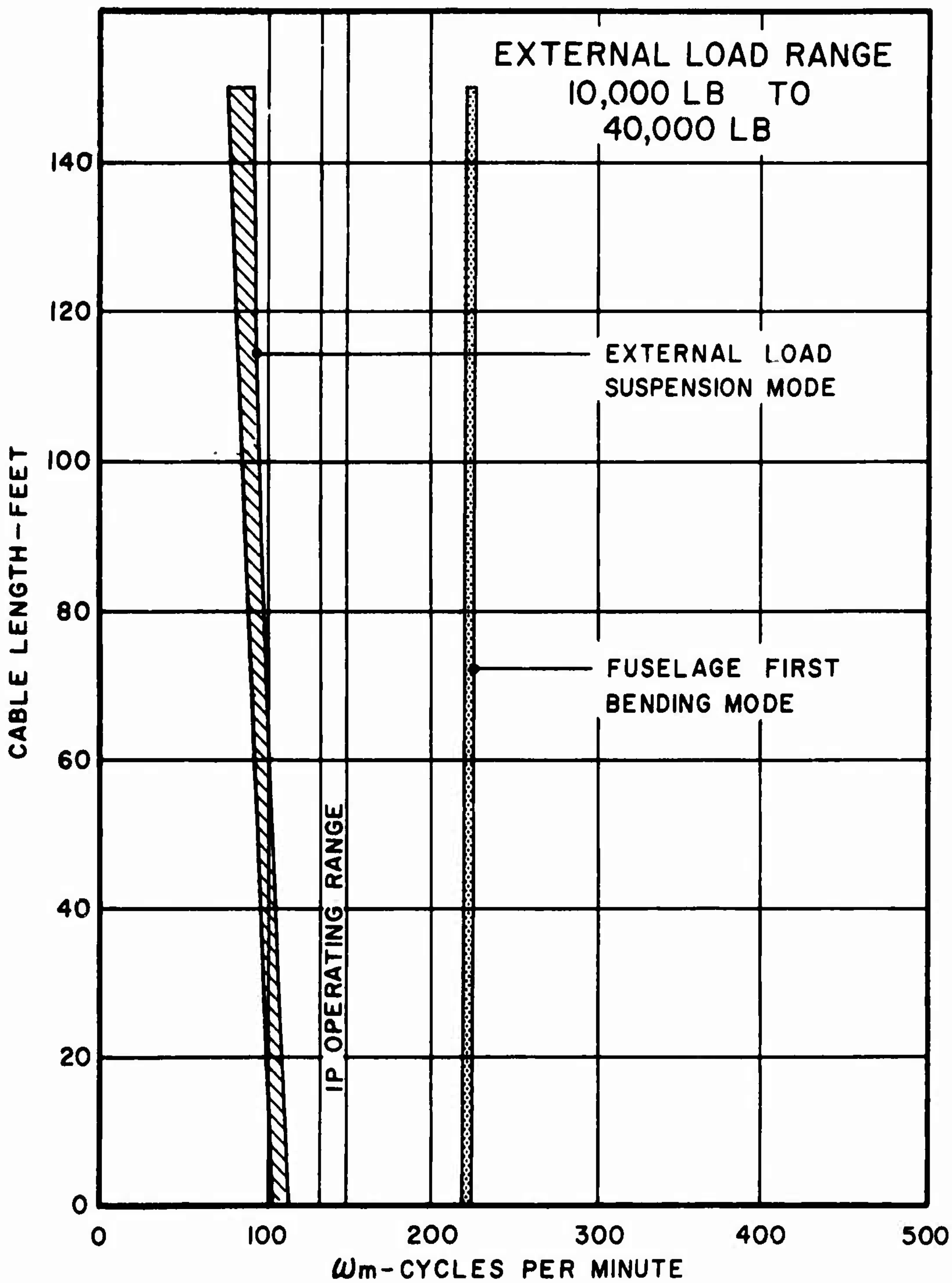


Figure 41. Vertical Bounce Mode Frequencies vs Cable Length With Decoupler.

on the single-rotor heavy lift helicopter. Whether the pod will clear the main landing gear is dependent on the landing gear design and aerodynamic forces and cannot be determined at this time. The interference effects between the pod and the fuselage are the major unknown factors and will have to be evaluated by wind tunnel testing before the use of in-flight jettisoning can be considered safe.

A pod used to carry personnel requires the use of fixed fuselage pod locks to ensure that accidental in-flight jettisoning cannot occur. If the pod is used to carry cargo, it is desirable to permit in-flight jettisoning. Thus it is necessary to include provisions in the locks for an explosive bolt release. The use of replaceable nonexplosive bolts when carrying personnel and explosive bolts when carrying cargo would introduce a serious human factors problem. For this reason the use of explosive bolts in the fuselage pod locks would be undesirable, hence precluding any possibility of in-flight jettisoning of the pod.

IN-FLIGHT ADJUSTMENT OF MULTI-POINT HOISTS

All multi-point systems have the capability of in-flight adjustment of one or more of the cable lengths. The advisability of making such in-flight adjustment at other than hovering or very low forward speed conditions is questionable. Preliminary analysis indicates that multi-point loads will assume a stable aerodynamic position for reasonably adjusted cable lengths at any given forward speed. In-flight changes in cable lengths may affect aircraft stability and tend to produce pitching oscillations. Further evaluation utilizing wind tunnel tests is desirable to obtain qualitative data upon which the limitations and/or advisability can be based.

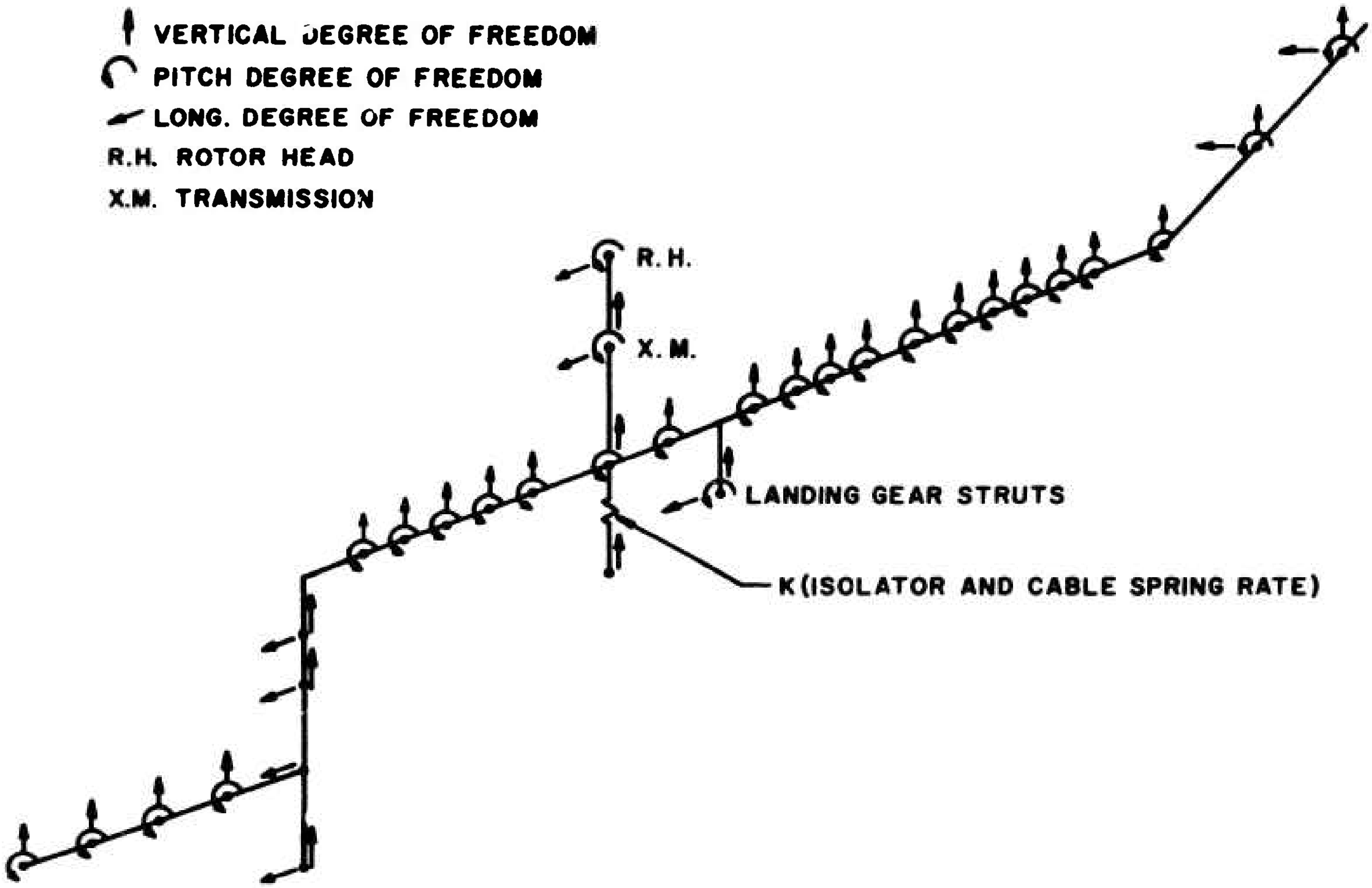


Figure 42. H.L.H. Fuselage Mathematical Model.

PROBLEM AREAS AND PROPOSED SOLUTIONS

MECHANICAL LOAD RELEASE FROM COCKPIT

One of the performance objectives of this study is to provide for two methods of cockpit controlled load release, electrical and mechanical. The mechanical release objective presents a major problem area.

One approach is the incorporation of a hydraulic line in the central core of the cable. The electrical conductors would then be used to replace one of the outer strands of the cable. The conductors would be suitably protected by wire braiding and would not support any of the loads. There would be no loss in strength of the cable, since, in the standard non-rotating construction, the outer layer of strands have substantially greater strength than the inner layer. Unfortunately, the size of the hydraulic line in the central core would, even with 3000 psi oil supply, give a relatively small output force. Leakage problems would have to be eliminated, and the added complexity required would probably negate the advantages expected by having a redundant hook release method.

Another possibility is the use of a push-pull mechanical cable in the core with the electrical conductors woven into the outer strands as described above. Unfortunately, the smallest available size of the cable is 3/8 diameter. Also, this type of cable does not lend itself to operation if it is forced to maintain a helical position. Considerable design and development work would have to be done in order to solve these two basic problems.

A third approach is the use of a mechanical release line supported on a separate, constant tension cable drum. A hydraulic or electrical motor drive would provide the tension required for hook release. The mechanical release line would be attached to the hook. The primary problem to be expected is that of this line winding around the primary load suspension cable. A secondary problem is that of the line becoming entangled in the equipment to be hoisted. An added weight penalty would also be incurred, and system complexity would be increased.

A fourth solution is the design of a cable strip-off feature. This feature requires incorporation of a clutch to let the load pull the cable off the drum. In addition, some protective device is required to prevent the free falling load from overspeeding the gear train. While this approach offers the most feasible method of providing a redundant method of load release, controllable from the cockpit, it results in the loss of both cable and hook. It also requires design of a clutch which can be released under full load.

Although it does meet the specific requirement for pilot operated mechanical hook release, a system utilizing radio control was also investigated. In this system radio signals are used to operate a battery powered release mechanism in the hook. Two or more separate and distinct radio signals must be used to trigger the hook release to prevent operation by random

radio signals. In addition, the hook batteries require regular recharging to ensure proper operation.

In summary, the development of a cockpit controlled mechanical hook release for any hoist system is considered to be a major problem area, whether or not it is integrated with the primary suspension cable. No problems exist with the design and fabrication of the cable sizes required for any of the hoists if only electrical conductors are required to provide the power for hook actuation.

Several of the configurations under study require a separate electrical conductor cable to provide the electrical power required to open the hook of the single-point hoist. Such a cable would be suitably protected by a braided wire jacket and wound on a reel. The use of a separate electrical conductor cable eliminates the need for conductors in the core of load suspension cables when a beam is used to convert from multi- to single-point hoist mode, as in the -5 and -16 configurations. It is more a matter of opinion than of fact that use of a separate electrical conductor cable offers advantages over that of conductors buried in, and suitably protected by, the load suspension cable.

Two alternate solutions for the requirement of a separate mechanical cockpit controlled load release are proposed.

The first alternative is simply reliance on the manual ground controlled hook release as the backup method of release. The use of a tandem-dual cable cutter controlled from the cockpit will serve as a secondary (emergency) release.

The other solution is to provide a cable strip-off feature in addition to the systems described above. Although this concept can result in loss of cable and hook (as well as load) if used under emergency conditions, it does provide a release mode entirely independent of the normal electrical release system for additional redundancy.

WEIGHT

All of the systems evaluated which offer separate functions for the single and multi-point hoists (the -1, -2, -3, -4, -11, and -13 configurations summarized in Tables XVII and XVIII, pages 87 and 89) weigh slightly more than the 4000-pound goal. However, the combined function systems (the -5, -6, -7, -14, -15, -17, and -18 configurations) will still provide an appreciable weight savings.

This separate function system weight penalty must be balanced against the redundancy offered by having two separate, independent systems available as well as the ability to reduce the aircraft empty weight for any specific mission by the removal of the major components of the system not to be used. This capability reduces the system weight well below 4000 pounds, and even below the weight of the combined function systems. All systems which require the combining of functions offer a total system weight below 4000 pounds. They offer no redundancy, however, and cannot be reduced in weight

to a noticeable degree by removal of major components.

SYNCHRONIZATION OF MULTI-POINT HOISTS

Synchronization of the cable travel (hook position) has been considered by several investigators to be a major problem area in a hydraulically powered multi-point hoist system. The inclusion of an electrical feedback system will limit the maximum variation in hook position to within 7 inches in a total cable excursion of 50 feet. In addition, the use of a relatively simple one-time check-out procedure will "zero out" most of the instrument error and further reduce the cable length variation to approximately 1-1/2 inches (see ERROR ANALYSIS, page 148).

COMPARATIVE RELIABILITY AND MAINTAINABILITY ANALYSIS

A study to compare the reliability, maintainability, and unavailability of the various configurations (single-point plus multi-point and multi-point alone) of the subject system has been made. Table XX gives the results of this preliminary study. The results are valid on a relative basis; however, on an absolute basis they are subject to considerable variability because of the limited data available at this time.

For each configuration considered, a failure or malfunction rate is given for both a single-point and a multiple-point mission. These rates are not to be interpreted as abort rates (unable to complete mission) but rather as rates of unscheduled maintenance actions. The failure rate for the single-point missions is the rate of malfunction expected for executing a single-point mission only. For example, with the -1 configuration, in 1000 hours of single-point mission flying, an average of 5.59 failures would be expected. Similarly, for the multi-point mission, the failure rate is associated with the type of mission only.

Also included in Table XX are comparative values for the maintenance man-hours per flight hour and unavailability for the various configurations. Unavailability is the complement of availability. These numbers are for the cargo handling system only.

Table XX also includes estimates of mission reliability data for the 13 external cargo handling system configurations. These data are presented in three columns as follows:

1. The column headed "Single-Point Only" gives the abort rate for the single-point system only. For example, in 1000 hours of single-point mission flying, the -1 configuration would experience an average of 1.86 aborts.

2. The column headed "Multi-Point Only" gives the abort rate for the multi-point system. For example, in 1000 hours of four point missions, the -1 configuration would experience an average of 3.46 aborts.

3. The column headed "Combined System" gives the abort rate for the cargo system as a whole for a 1.76-hour and a .463-hour mission. This assumes that either the single-point or the multi-point system could be used for any mission. Where there is no redundancy, the lowest abort rate for single or multi-point suspension is applicable.

TABLE XX

RELIABILITY/MAINTAINABILITY COMPARISON,
H.L.H. EXTERNAL CARGO HANDLING SYSTEMS

Config-uration	Failure Rate: Failures, 10^3 Hrs. Single Point	Failure Rate: Failures, 10^3 Hrs. Multi-Point	MMF/FH	Unavail.	Failures $/10^3$ Hrs.	Mission Reliability Aborts/10^3 Hrs.: Single Point	Mission Reliability Aborts/10^3 Hrs.: Multi-Point Only	Combined System 1.76 Hr. Mission	Combined System .463 Hr. Mission
-1	5.59	19.91	.1936	.0036	25.50	1.86	3.46	.01135	.00298
-2	7.17	19.24	.1882	.0038	24.77	1.24	3.34	.290	.286
-3	5.74	14.88	.1508	.0018	19.17	1.91	4.95	.481	.473
-4	5.74	21.86	.1955	.0028	26.34	1.91	5.56	.483	.473
-5	22.79	21.64	.1755	.0035	22.79	3.96	3.78	3.78	3.78
-6	27.11	35.91	.2661	.0058	35.91	4.70	6.23	4.70	4.70
-7	24.11	23.11	.1842	.0037	18.89	4.18	4.01	4.01	4.01
-11	5.74	10.44	.1305	.0027	16.18	1.91	1.81	.00608	.00161
-13	5.67	8.10	.1030	.0012	12.43	1.89	2.70	.451	.446
-14	7.36	17.07	.1514	.0027	19.07	2.46	3.82	2.46	2.46
-15	15.22	12.87	.1235	.0023	15.22	2.64	2.23	2.23	2.23
-17	14.47	12.47	.1181	.0022	14.47	2.51	2.16	2.16	2.16
-18	21.49	9.62	.1675	.0029	21.49	5.59	3.21	3.21	3.21

EVALUATION PROCEDURE

INTRODUCTION

As an aid in selecting the external cargo handling system configuration that best meets the 40,000-pound payload requirements of the heavy lift helicopter, an evaluation procedure employing both qualitative and quantitative factors pertinent to the system design was employed.

Productivities or costs are calculated for each of the competitive cargo handling designs. These results are combined with qualitative factors and are evaluated in a comparison matrix. The matrix attempts to quantify the qualitative factors in relation to productivity or cost by assigning relative weighting values to each of the parameters. These relative weighting values are based on judgment. Each of the cargo handling subsystems is then scored. Systems within 5 pct to 10 pct of each other are considered to be equivalent.

DISCUSSION

This section presents a discussion of the methodology used by Sikorsky Aircraft to select an optimum cargo handling subsystem for the heavy lift helicopter system. The classic steps in devising a good selection process are:

1. Acquire complete understanding of the operational concept and the functional requirements of the system and subsystem.

2. Establish criteria of effectiveness and/or cost as a basis for system selection.

3. Identify all relevant factors that are pertinent to the cost and effectiveness of the operation.

4. Classify all relevant factors into qualitative and quantitative categories.

5. Quantify all the factors that are possible.

6. Construct a mathematical simulation model relating all quantifiable factors to the criteria of selection. This model could consist of a simple equation or it could consist of a complex set of equations requiring computer processing.

7. Exercise the simulation model to determine effectiveness and/or cost.

8. Examine all qualitative factors.

9. Evaluate and relate all factors and select optimum system.

One of the most important steps in the evaluation process is to acquire complete understanding of the system operational concepts and mission requirements. A comprehensive missions analysis is required to develop the technical requirements for the heavy lift helicopter and its cargo handling subsystem. These missions and operational analyses are presently in progress at Sikorsky Aircraft, but as in all concepts, such analyses are difficult to perform because of the uncertainty of so many of the basic parameters. Some of the more complex factors of such an analysis include consideration of the types of loads to be transported and their frequency distribution, frequency distribution of operating ranges, new loads packaging concepts and methods of suspension, logistics and maintenance problems of field conversion of single-point and multi-point suspension equipment, etc.

These factors are difficult to define at this time because the crane concept is still relatively new. Even though the U.S. Army has had over 2 years of field experience with the CH-54A in the Continental U.S. and in S. Viet Nam, new operational concepts are formulated almost daily. This is complicated by the fact that the heavy lift helicopter configuration has not been fully defined.

For the purposes of this study, the mission profiles described in the BASIC DATA section (pages 3 and 4) were used, assuming equal distribution of single-point and multi-point suspensions loads and equal distribution of long and short range operation. Future missions and operational analyses may modify these assumptions, and therefore changes to some of the conclusions of the evaluation may result.

The criteria of effectiveness used to evaluate the cargo handling subsystem can be either productivity of the heavy lift helicopter crane system or total system cost. The effect of the cargo handling subsystem on the performance or cost of the total heavy lift helicopter crane system is important, not the absolute differences of values for the cargo handling equipment alone. For example, one cargo handling subsystem may be more reliable but heavier than another subsystem. The more reliable subsystem may not be the better one, since it is possible that the gain in system productivity due to the higher reliability may be more than offset by the loss in productivity due to the increased weight.

Relevant factors that are considered in the evaluation process include weight, reliability, power required, maintainability, stability, safety, load acquisition and release time, design compromise to the heavy lift helicopter airframe, versatility, logistics problems due to conversion, development problems, and cost. Most of these attributes can be quantified into productivity or cost relationships. However, factors such as versatility, airframe design compromise, and logistics problems are non-quantifiable at this time. These attributes will be analyzed qualitatively and will be considered on the basis of past experience and good judgment.

DESIGN OBJECTIVES

As mentioned earlier, in order to evaluate the various cargo handling subsystems effectively, all factors pertinent to the performance effectiveness and cost of the operations must be identified. The interrelationships and the sensitivities of these factors are determined by combining them into a single effectiveness parameter of either system cost or productivity (ton-miles/hour). It was felt that system productivity was as accurate as cost and was much easier to use, so productivity was selected as the criterion of effectiveness.

A discussion of the relevant factors is presented in the following paragraphs to ensure that quantitative ratings of these parameters as used in the productivity formula are considered in their proper perspective and are not accepted as unequivocal ratings.

System Weight

Calculations have been made to determine the overall system and special mission weights for each of the alternative hoist systems under consideration. The mission weight is defined as the weight of equipment required to accomplish a specific mission, such as a single-point load, and is lower than the system weight in most configurations because major components of the single- or multi-point system could be removed prior to performing a mission.

There are several other design considerations which must be taken into account while minimizing system weight. For example, a design for minimum weight might not be as satisfactory as a design which permits interchangeability of moving parts because of the savings in maintainability and logistics. Extra design features for ease of maintainability such as accessibility or quick disconnects can become more important than an associated weight penalty.

System weight plays an important part in the determination of the productivity of a configuration. Although each alternative hoist system is designed for a 20-ton capability, the aircraft performance is penalized by system weight. This penalty may affect system effectiveness in terms of reduced productivity if a constant gross weight heavy lift helicopter is assumed; or if the gross weight is increased to maintain a 20-ton payload, the penalty will be increased procurement cost.

Another consideration which should be taken into account when calculating system weight is the type of supporting structure within the airframe. The single- and the tandem-rotor aircraft configuration will be affected in different ways by the many possible hoist system configurations. External placement of the motors, pumps, and hoists will affect the drag, while internal placement will disrupt the location of fuel cells and passenger or cargo compartments; thereby, additional structural weight penalties will be required.

These comments have been made to show that the best system is not necessarily the one having the lowest system weight. There are structural weights associated with each system which have only been estimated. In addition, there are other design factors which demand a degree of sophistication resulting in slightly increased weight.

Safety

The factor of safety is one which must be expressed in qualitative terms rather than in quantitative terms. The design of a system may be reviewed and rated as either safe or unsafe. Naturally, those criteria responsible for an unsafe rating must be corrected before the design is acceptable.

One of the most critical factors considered within the category of safety is the stability of the load. As the aircraft speed increases, the load stability changes and this in turn affects the controllability of the aircraft. Speed is one of the more sensitive items in the productivity study, so maximum speed allowable within the limits of safety is desirable, and that system which allows the greatest speed within safety limitations is regarded as the best.

Each of the suspension systems is capable of carrying any type of load by using various combinations of bridles. However, the manner in which the load is supported has a direct bearing upon its in-flight stability. Oscillations of a load affect control of the helicopter because of the changing drag factor and the change in location of the center of gravity of the load. Suspension of a load by a four-point system provides the most stable means of support and hence allows the greatest forward airspeed. Precise control of the load during flight permits even greater load stability throughout the changing attitudes of the aircraft. This can provide the safest system capable of the highest airspeed. However, load stability is also dependent upon its density. High density loads can be carried satisfactorily from a single-point system; high drag-low density loads such as downed aircraft cannot be stabilized even through the use of multi-point hoists, and drogue chutes are required to prevent in-flight oscillations. In these cases, safety of flight demands low airspeed; hence, productivity is a poor measure for comparing alternate hoist configurations.

Reliability

Reliability is generally regarded as the likelihood that a given system will function normally. With a system as sophisticated and expensive as a heavy lift helicopter, it is important that the cargo handling subsystem be as reliable as possible. Failure of the cargo handling system could result in grounding the entire helicopter system.

Reliability is measured in terms of failure rates or unscheduled maintenance requirements. Thus, a multi-point system would be considered less reliable because, with a larger number of components, there are more chances that something will fail. A review of the type of failures should be made because many would occur to those components which have a direct

association with the raising and lowering of a load. If this is true, it might still be possible to lock the cable drums in position and utilize the extended cable or cables as a form of bridle. This would allow utilization of the helicopter to perform a mission with performance somewhat decreased due to load instability.

In several of the configurations studied there are duplicate systems, single and multi-point, so that failure of one system would not necessarily result in an aborted mission. Through the use of bridles or slings, both the single and multi-point systems can handle all types of loads. Thus it is necessary to evaluate the penalty of greater system weight and higher maintainability of both a single and multi-point system against higher mission reliability.

Maintainability

The factor of maintainability is most easily represented by cost. The maintenance man-hours per flight hour for each system can be measured and converted into dollars by using standard labor rates. However, when the factor of maintainability is to be included in the productivity formula, it is represented as a function of availability. The availability of the heavy lift helicopter is penalized by the downtime due to maintenance. This may not be completely true, however, because the scheduled maintenance and sometimes unscheduled maintenance of the cargo handling system can be done simultaneously with maintenance of the aircraft itself. The fallacy of this method is that in comparing two closely rated systems, the factor of time attributed to maintainability receives the same emphasis as other time factors such as loading time which have a direct effect on mission accomplishment.

There are other factors to be considered in the area of maintainability which may not appear in the availability figures. Standardization and commonality of parts is important when considering the storage and handling of spares. Accessibility and vulnerability of components is important, particularly for those components which may be removed for weight saving purposes for daily missions. Check-out of the system without starting the main engines and rotor blades is important, since it permits the check-out to be completed by the ground crew instead of the pilot. Characteristics of this type are not always taken into account by quantitative measures and as availability, but they should be considered in the selection of the recommended system.

Size

The physical size of the systems does not appear in the productivity analysis because it does not directly affect the size or weight of cargo which can be transported. It is important to consider the physical size of the system when considering its installation within an airframe. Size is less significant when the system is to be mounted on a crane type helicopter than if it is to be mounted within an existing cabin. It is assumed that the configurations will all be competitive in size and that it would be feasible to install any of the systems in an aircraft.

Power Requirement

It is difficult to determine the sensitivity of the power required for several alternative systems, especially in a productivity model. If the basis of comparison is cost, dollars per horsepower are readily assigned and provide a relative rating. If the comparison is to be made on the basis of productivity, differential power required would be translated into pounds per horsepower and the total pounds would be subtracted from payload.

System Cost

Initial system cost is considered to be important but by itself can be very misleading. Not only should lifetime cost be considered, but logistic cost due to maintainability and reliability must be considered. All of the above-mentioned design objectives can be measured as a function of cost and can be related to the cost of the overall system operation. In addition, there are several other qualitative factors which are not easily assigned a dollar figure but do affect operating costs of the system.

Qualitative Selection Factors

The preceding paragraphs discussed the design objectives assigned in the cargo handling work statement provided by the Army. There are other design criteria which may be considered of less significance but which should have some bearing on the system selection. Examples of some of these are as follows:

Ground Handling Time

Minimum ground handling time is important during load attachment because it has a direct effect on helicopter hover time. If the assumption is made that this system would be used to carry Army Division equipment between 10 and 20 tons, 30 pct of that equipment must be attached while the helicopter is in a hovering position. Hookup time then becomes a factor which could influence the choice of a system. The time required for ground preparation of the load, such as the bridle attachment prior to arrival of the aircraft, should not be included. This time should perhaps be included with maintainability, since it directly affects availability.

Versatility

The current study has shown that the one- , two- , and four-point systems are each capable of handling any type load through different combinations of bridles. The degree of effectiveness with which the different systems can handle all loads is not the same. Therefore, further information on the type of loads to be carried could also influence the selection of a system.

The size of pods to be used for hospitals, troop transports, or command posts should be standardized in height and width and perhaps in multiples of length. These could then be transferred to or from railroad and highway vehicles. As the super transports such as the C-5A become a reality, much Army equipment will be packaged in containers being specially designed for efficient loading and unloading. These containers will remain intact from shipper to destination so that compatibility with a helicopter handling system is important. Similarly, there is growing interest in the Fast Deployment Logistic (FDL) and in programs which include Amphibious Helicopter Assault Ship (LHA) plans for unloading ships by helicopter.

In addition to the pod or container, pallets must be accommodated for aerial flight. Pallets can be used not only for loading small-sized containers but for transporting multiple vehicles or trailers with their associated towing vehicles. Power suspension is extremely important in these situations to prevent loss of load during flight.

Another element of versatility to be considered in selection of a system depends upon the desirability of splitting loads among several destinations. A multi-point system could be used to make more than one drop without requiring assistance from the ground for rearranging bridles or making other adjustments. Limitations for this type of operation are established by helicopter center-of-gravity limitations and by the weight of the cargo being carried.

Human Factors

After considering the advantages of multi-point systems due to the greater flexibility in manipulating the load, investigations should be made to determine that the pilot or other crewman has the means of controlling the load. This includes means of knowing the position of the load and individual tensions on the cables. Knowing this information is only part of the problem. The remainder of the problem is providing the means of independently operating the cables so as to reposition the load. Even with adequate visual facilities to sight the load, displays are required in the aircraft due to the difficulty of depth perception when the ends of the cables are more than 20 feet below the aircraft.

Structural Design Compromises

At the outset of the study, the philosophy was to define a cargo handling system without reference to a particular type of airframe. It was felt that this would ensure the best type of cargo system which then could be attached to any type of airframe. As different rating methods were tried and different mission profiles examined, no single factor or group of factors seemed to give any of the alternative systems significant advantages. As various criteria were considered, the impact of the method of installation within an airframe became more relevant. The effect of system weight was briefly mentioned but there are other considerations in both the

single- and tandem-rotor configurations which could influence selection of a system. In addition to the structural arrangement of these types of helicopters, each has inherent handling characteristics which will tend to influence the relative merits of the systems. The criteria of compatibility should be considered at least as strongly as the other qualitative criteria.

Productivity Analysis

Productivity is the rate of payload delivery for a given mission. This is the major quantitative parameter that will be used to compare the performance effectiveness of the various cargo handling designs. The best configuration enables the heavy lift helicopter to deliver the greatest amount of payload over a specified distance in the shortest period of time. Productivity is a function of payload, radius of operation, availability, mission reliability, and total time to accomplish the mission. The weight differences of the various cargo handling designs are accounted for by modifying the payload to penalize any cargo handling design that is heavier than the base subsystem. The reliability and maintainability of the equipment are reflected in availability and in mission reliability. Load stability are reflected in cruise speed, acceleration, and deceleration of the heavy lift helicopter. Complexity of cargo hookup and release is reflected in loading and unloading time. The equations and the assumptions used to determine productivity are described below:

$$\text{Productivity} = \frac{\text{Payload x Distance x Availability x Mis. Rel.}}{t} \quad (24)$$

where

Payload = capacity of the aircraft less the difference in weight between the configuration in question and the lightest configuration (lightest configuration has 20-ton payload).

Avail. = assumed base availability of the aircraft less difference in unavailability between configuration in question and least available configuration. Assume aircraft has .85 availability.

Mis. Rel. = percentage of missions which, once initial, will not be aborted due to cargo handling system failure.

$$t = t_9 + n\,(t_7 + t_1 + t_3 + t_5 + t_8 + t_2 + t_4 + t_6) \quad (25)$$

n = number of round trips

t_1 = time to accelerate to cruise speed with load

t_2 = time to accelerate to cruise speed without load

t_3 = time at cruise speed with load

t_4 = time at cruise speed without load

t_5 = time to decelerate with load

t_6 = time to decelerate without load

t_7 = time to load

t_8 = time to unload

t_9 = time to warm up

The cargo handling systems have been evaluated for single and multi-point cargo suspension for the following missions:

12-ton transport: One round trip of 100-nautical-mile radius carrying 12 tons outbound only.

20-ton transport: One round trip of 20-nautical-mile radius carrying 20 tons outbound only.

The assumptions made in the productivity analysis are shown in tabular form:

	1 + 4 Pt.	4 Pt.	1 + 2 Pt.	2 Pt.
Time to warm up, min	3	3	3	3
Acceleration without load, n.m./sec	4	4	4	4
Cruise speed without load, kn	130	130	130	130
Angle of climb	30°	30°	30°	30°
Deceleration without load, n.m/sec	4	4	4	4
Hoist speed, fpm (lower and raise 20 ft)	60	30	60	30
Time to load, min	1/2	2	1/2	1
Acceleration without load, n.m./sec	3	3	3	3
Cruise speed with load, kn	60	60	60	60
Deceleration with load, n.m./sec	3	3	3	3

	1 + 4 Pt.	4 Pt.	1 + 2 Pt.	2 Pt.
Time to unload, min	1/2	2	1/2	1

Note: For loads carried single point and assuming the hookup is made with the least number of hooks available.

The same productivity formula applies for loads carried on the multi-point suspension and assuming the hookup is made with the largest number of hooks available with the changes in assumptions listed below:

	1 + 4 Pt.	4 Pt.	1 + 2 Pt.	2 Pt.
Hoist speed, fpm	30	30	30	30
Time to load, min	2	2	1	1
Cruise speed with load, kn	100	100	80	80
Time to unload, min	2	2	1	1

The results of the productivity calculations for each configuration are given in Table XXI, page 129. An average productivity for these data is also presented.

QUALITATIVE EVALUATION

As has been previously indicated, many of the factors governing the selection of an external cargo handling system cannot accurately be quantified and included in the previous evaluation method. These factors must therefore be ranked solely on the basis of judgment and past experience. These factors are then combined with productivity in a comparison matrix to select the best system.

It is difficult at this point in the study to establish accurate cost information. Therefore, to simplify preparation and evaluation, the 13 configurations have been given a relative ranking for prototype hardware and development costs. The following factors are included in the qualitative comparison matrix:

FACTOR	RELATIVE WEIGHTING
Productivity	20
Safety	10
Cost & development effort	5
Versatility	5

FACTOR	RELATIVE WEIGHTING
Airframe compatibility	5

Productivity is given a weighting of 20 because it is a function of weight, reliability, and maintainability, as well as load stability in flight and acquisition time.

Safety, which includes evaluation of potential safety hazards in all operational phases including load acquisition, flight, and load release has been given a weighting of 10. In ranking the various configurations, the single-point load suspension has been considered the safest for flight conditions, since triple redundancy is provided (tandem-dual cable cutters and cable strip-off) for emergency jettisoning. Two-point and four-point suspensions are considered to have approximately equal safety characteristics. While the four-point has a lower overall cable cutter reliability than the two-point, a four-cable suspension is considered to offer sufficient redundancy to offset this factor.

The cost of prototype hardware and development effort, versatility, and the compatibility of the system to the airframe have been given weighting factors of 5.

A summary of the results of the qualitative comparison matrix for the heavy lift helicopter is presented in Table XXII, page 130. Configurations with scores within 5 pct of each other are considered to be approximately equivalent.

TABLE XXI

SUMMARY, PRODUCTIVITY ANALYSIS

Mission	Suspension	Configuration Productivity (Ton n.m./Hr.) 1	2	3	4	5	6	7	11	13	14	15	17	18
12-Ton Transport	Single-Point	400	401	402	402	368	379	381	397	403	394	384	389	387
	Multi-Point	487	488	490	484	501	514	524	430	435	463	447	466	463
20-Ton Heavy Lift	Single-Point	587	588	589	589	516	520	523	586	591	569	550	554	552
	Multi-Point	653	655	656	651	648	638	643	636	645	633	636	636	633
Average Productivity		532	533	534	532	508	513	518	512	519	515	504	511	509
Relative Productivity		9.9	9.9	10.0	9.9	9.5	9.6	9.7	9.6	9.7	9.6	9.5	9.6	9.5

TABLE XXII
HEAVY LIFT HELICOPTER
QUALITATIVE EVALUATION MATRIX

	Weighting Factor	Configuration 1	2	3	4	5	6	7	11	13	14	15	17	18 Total Value*
Productivity	20	198	198	200	198	190	192	194	192	194	192	190	192	190
Safety	10	80	80	80	50	60	60	60	80	80	70	60	60	60
Cost & Development	5	40	40	35	20	45	40	50	40	35	30	50	40	40
Versatility	5	50	50	45	30	35	35	35	45	35	30	30	30	30
Airframe Compatibility	5	45	50	30	20	50	50	50	35	25	30	40	30	30
Total		413	418	390	318	380	377	389	392	369	352	370	352	350

*Note: Each configuration was ranked from 1 to 10 for all qualitative factors. The total index value for each factor was obtained by multiplying the weighting factors and the ranking.

SUMMARY

All 13 configurations studied during Phase I have been evaluated with respect to the following parameters:

1. Weight
2. Reliability
3. Safety
4. Maintainability
5. Physical Size
6. Power
7. System cost
8. System development effort
9. Technical confidence
10. Redundancy
11. Versatility
12. Airframe compatibility

On the basis of this Phase I investigation, an external cargo handling system incorporating a mechanically powered single-point hoist and hydraulically powered multi-point hoists is considered to best meet the requirements of a 40,000-pound-payload heavy lift helicopter.

For a single- plus four-point system the -1 configuration is recommended.

For a single- plus two-point system the -11 configuration is recommended.

The major factors leading to this selection are as follows:

1. While the -1 and -11 separate function configurations are heavier than the combined function systems and exceed the 4000-pound goal by approximately 10 pct as a total system, the capability of major component removal results in a single function weight of approximately 60 pct of the design goal.

2. Separate function systems such as the -1 and -11 configurations provide better aerial cargo system availability, since the mal-

function of one system does not necessarily down the aircraft. The single and multi-point systems of these two configurations have entirely separate power sources. This provides greater drive system reliability than that of the separate function systems.

3. A single-point load suspension is considered the safest for flight conditions, since the normal electrical release mode is backed up by an emergency system with triple redundancy. Even total malfunction of all release modes does not inherently produce aircraft instabilities, as in the case of partial failure of any of the multi-point systems combined to perform single-point missions.

4. The -1 and -11 configurations require the least development effort to achieve the desired goals of this program. In addition, the proposed clutch-reverser unit for the mechanical drive single-point hoist can be replaced by either the variable speed drive unit or a hydraulic motor if these systems prove to be advantageous as the state of the art is advanced.

5. The -1 configuration was selected rather than the -2 primarily because it utilizes different power sources for the single- and the multi-point hoist systems. A hydraulic motor, pump, or clutch failure would not prevent use of the aircraft for a cargo handling mission. In addition, both the pump and the motor proposed for the -2 configuration require further developmental effort.

6. The -1 and -11 configurations can be adapted to either the single- or the tandem-rotor aircraft. They are easily adapted to the single-rotor type because of the proximity of the accessory gearbox and the single-point hoist. Addition of a gearbox between the drive shafting connecting the forward and aft rotors is required to provide a power source for the clutch-reverser unit on the tandem-rotor aircraft.

The -1 configuration utilizing a single-point hoist and four multi-point hoists is considered the better of the two, especially for the single-rotor aircraft, because of its versatility and compatibility with airframe and structure.

1. For short-range missions, the single-point load suspension is the most versatile. It requires lower hookup and release times and has higher hoisting speeds. Analysis indicates that the four-point system has only a slightly better aircraft cruise speed advantage over the two-point system for missions of longer range. However, the four-point load suspension has better versatility, since no special bridle or rigging is required to carry a wide variety of vehicles, as is the case for any two-point system.

2. The -1 configuration is more compatible with the airframe structure, since it requires only one well for the single-point hoist installation. The single-point hoist is particularly adaptable to the single-rotor helicopter, since it is located directly under and utilizes the main transmission support structure. The four-point hoists are located outboard of the fuselage structure on davit type structures with suitable aerodynamic fairings.

3. The -11 configuration with a single- plus two-point load suspension requires three airframe wells. In addition to a small airframe weight penalty, these wells utilize the crane-type fuselage cavities normally used for fuel tankage.

PHASE II

PRELIMINARY DESIGN

DISCUSSION

The external cargo handling system selected for the preliminary design phase of this study is a single- plus four-point arrangement utilizing a mechanically driven single-point hoist and four hydraulically driven zero-moment hoists as shown in Figures 68 and 69, pages 251 and 253.

The single-point hoist is capable of raising and lowering a 40,000-pound load 100 feet at a rate of 60 feet per minute. The single-point hoist drive train consists of a clutch-reverser unit, two angle gearboxes, and several sections of shafting. On the single-rotor heavy lift helicopter, the clutch-reverser unit is mounted on the accessory gearbox (Reference 3, Figure 58, page 245) permitting operation on the ground from the auxiliary power plant with the rotors stationary as well as from the rotor system in flight.

The four-point hoists are rated at 11,550 pounds at a speed of 30 feet per minute and contain 50 usable feet of cable. These units are hydraulically driven by a pump mounted on the accessory gearbox and utilize a hydro electric feedback control system to ensure synchronized load lifting. The four-point hoists can be operated on the ground without the rotors turning when the accessory gearbox is driven by the auxiliary power plant.

Both the single-point and the four-point hoists are readily removable from the aircraft when missions requiring minimum empty weight are to be undertaken.

For the tandem-rotor heavy lift helicopter, the single-point hoist has been located with the drum axis parallel to the aircraft longitudinal centerline (B.L. 0 as shown in Figure 70, page 255). This arrangement was selected to accommodate the somewhat smaller lateral trim moment capability of the tandem. Since the hoist is driven from the interconnecting shafting, it cannot be operated from the APP on the ground with the rotors locked.

The investigation conducted herein, as well as the experience gained on the CH-54A Skycrane, has clearly demonstrated to the authors that the basic preliminary design of any external cargo handling system is interrelated and should be conducted concurrently with that of the basic airframe. This is particularly true for the mechanically driven single-point hoist system selected. This system is undoubtedly more suited to the single-rotor heavy lift crane type helicopter (where the authors' experience lies) than to the tandem. Several other single-point hoist designs, although somewhat more complex, might be more appropriate for the tandem in view of its lateral trim limitations. Included among these are:

Large diameter, narrow width multi wrap drum

Capstan type, zero-moment

Traversing drum

Two part, doubled reeved

If rotor locked ground operation is mandatory, the use of hydraulic power will most likely result in an appreciable weight savings at a slightly lower mission reliability (see Table XX, page 117).

The hydraulically powered four-point hoist system is equally adaptable to both tandem- and single-rotor helicopters. The structural and hydraulic system design problems associated with hoist mounting drive and control are similar for both aircraft. The four-point hoist installation for the tandem-rotor heavy lift helicopter is shown in Figure 70, page 255.

SINGLE-POINT HOIST SYSTEM

HOIST

The single-point hoist, Figure 71, page 257, is a one-part, single-reeved type. The cable is wrapped in a single layer on a grooved drum, which is coated with a 0.25-inch-thick polyurethane rubber jacket and is attached to one end of the drum by a pin and clamp type fitting. Even winding of the cable on the drum is maintained by a level wind mechanism consisting of a ball screw driven bellmouth assembly. The level wind structure is free to rotate about the drum axis so that towing loads, with the cable swing 60° aft, will not result in high loads being reacted through it. In addition, the level wind contains scrub rollers which provide the power required to pull the cable off the drum without the need for an added weight on the hook. The scrub rollers are adjustable to compensate for wear.

Limit switches are also mounted on the level wind. All gearing required to power the ball screw and scrub rollers is supported in one level wind support arm. The bellmouth assembly is mounted on the ball screw nut and a reaction pipe. The reaction pipe provides the restraint required to transmit the rotary action of the ball screw into linear motion of the bellmouth-nut assembly. It also reacts the moment created when lateral cable swing is encountered. The bellmouth assembly contains the cable cutters which are used to shear the cable in the event of an emergency. A split, easily removable wear liner is also contained in the bellmouth assembly.

A Weston brake (Reference 6) is provided to prevent the load from dropping in the event of a power failure. It functions automatically and is located between the first and second stages of the hoist reduction gearing.

A hydraulically powered free reel clutch assembly is built into the hoist gearbox to permit the hook and cable assembly to be jettisoned in the event of a hook release malfunction. A manual control valve actuated by a push-pull control cable from the cockpit diverts flow from the aircraft utility system to the free reel clutch assembly. It is reusable and resets automatically when the hydraulic pressure is removed. The Weston brake is located between the input shaft and the free reel clutch assembly so that it does not affect proper free reeling operation.

A completely sealed slip ring assembly is provided to permit transfer of electrical control and power signals to the electrical conductors in the core of the main suspension cable from the aircraft's electrical system. A quick disconnect is provided to permit field removal of the cable without affecting the slip ring assembly.

A conventional gear reduction system with an overall reduction ratio of 443 to 1 is used. The first and second reduction stages of 4.190 to 1 and 3.513 to 1 are conventional spur gears. The third stage of reduction, 30.111 to 1, is a compound planetary system. All internal gearing is

splash lubricated and completely sealed. Oil fill, drain, and level plugs are provided.

Auxiliary drives are conventional spur gears. These gears are coated with a dry film lubricant and in addition are lightly coated with grease. The gears are enclosed by weathertight fiber glass shields which are easily removable for inspection and servicing. A shear joint is provided to protect the ball screw assembly in the event that excessive side loads should tend to jam the level wind assembly. Replacement of the shear pin, following the removal of the cause of jamming, is possible without disassembly of the hoist.

A cable length potentiometer is provided. It is easily accessible both for maintenance and/or replacement. An anti-backlash cover is provided to prevent the cable from jumping off the drum in the event a load is air dropped.

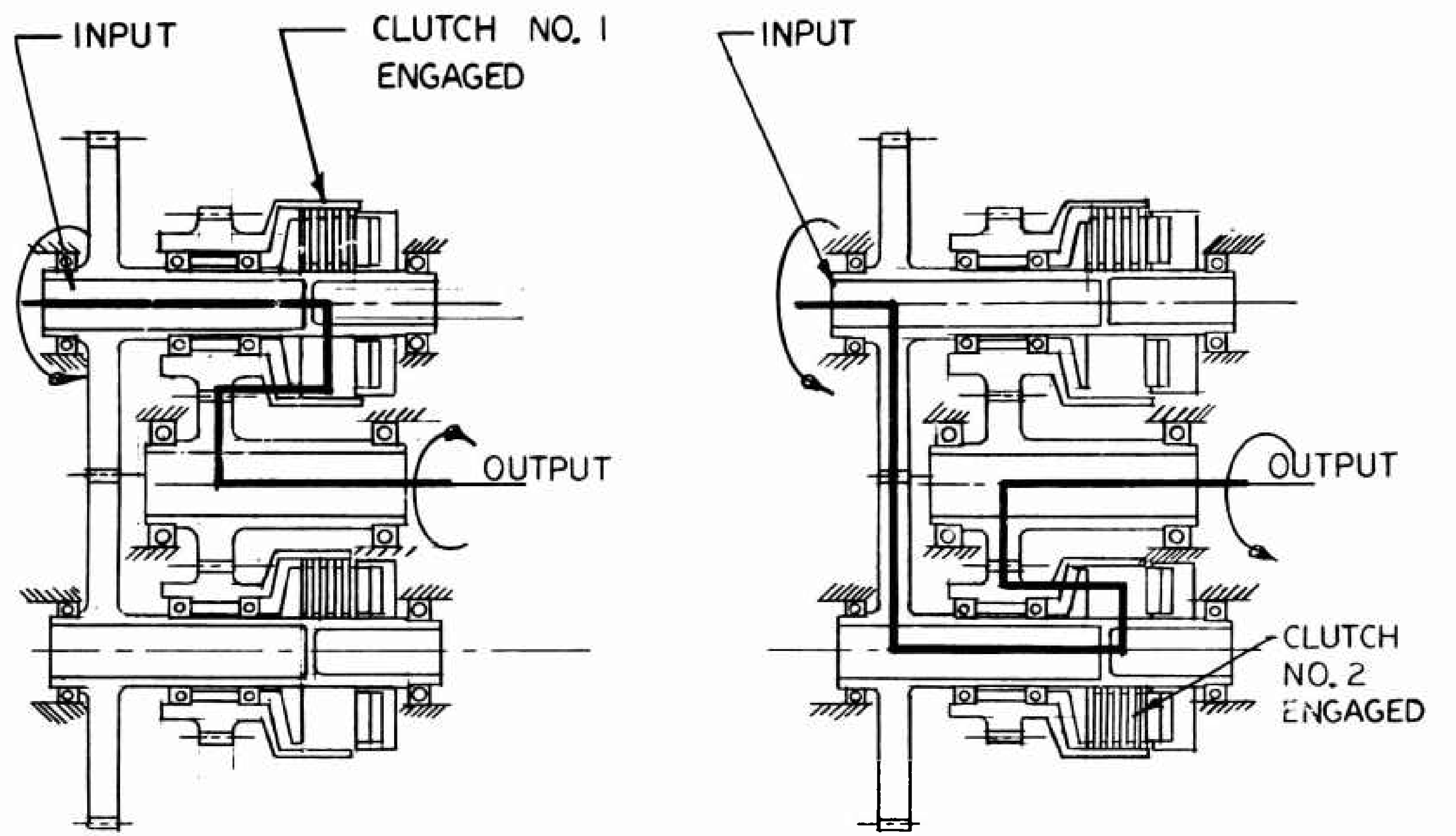
CLUTCH-REVERSER UNIT

A clutch-reverser unit mounted on the accessory drive gearbox provides the power to operate the main hoist. It is a conventional reversing gear similar to the type commonly used in marine applications. The unit consists of five spur gears, two oil actuated wet plate clutches, and a bevel gear set as shown in Figure 72, page 259. The oil used to engage the clutches and to provide clutch plate coolant when the hoist is inoperative is supplied by a gear pump through appropriate solenoid operated valves. This clutch control-lubrication pump is mounted on the accessory gearbox.

The clutches are spring loaded in the disengaged position. To hoist the load, oil at 250 psi pressure is supplied to clutch number 1. Clutch number 2 remains disengaged. To lower, the sequence is reversed. When hoist operation is not required, both clutches numbers 1 and 2 are disengaged. The hoist drive shafting is therefore stationary except when loads are being raised or lowered. Smooth clutch engagement is attained by a modified form of pressure modulation accomplished by use of an orifice between the accelerator and force cavities.

When the clutch is engaged, pressure oil enters the accelerator cavity. Since the accelerator cavity displacement is small, the pressure drop is momentary. The clamping force is then generated by a controlled pressure buildup in the force cavity created by the metering of a small amount of oil required through the orifice in the accelerator piston. A schematic showing this operation is shown in Figure 43.

The mechanical variable speed drive unit of Appendix II, page 245, was investigated as an alternate means of providing power to and reversing direction of the single-point hoist.



CLUTCH NO. 1 ENGAGED-HOIST RAISING

CLUTCH NO. 2 ENGAGED-HOIST LOWERING

Figure 43. Clutch-Reverser Unit Operation.

CABLE

The single-point hoist cable is stainless steel and is of 18 x 19 non-rotating construction. It has an outside diameter of 1.39 inches and a guaranteed minimum breaking strength of 150,000 pounds. The individual wires are .060 inch in diameter and are made from heat treated type 302 stainless steel. The 18 x 19 construction, 18 strands with 19 wires per strand, gives a flexibility greater than that attained in the 18 x 7 cable presently used in the CH-54A main cargo hoist.

The core of the cable contains seven electrical conductors of stranded wire construction. They are helically wound and encased in a tough, resilient, plastic jacket. Five of these conductors are required to transmit the electrical power required to operate the actuating solenoid in the cargo hook. Two conductors are spares; thus, field level maintenance personnel can "wire out" defective conductors without removing cable. This "wiring out" is possible by using the following procedure:

(a) Remove hook from swivel assembly and slip ring assembly from swivel.

(b) Remove slip ring assembly from hoist by pulling only far enough out of housing to service.

(c) Check continuity to determine good conductor.

(d) Connect good conductor to proper terminals after removing defective conductor.

(e) Reinstall slip ring in hoist.

(f) Reinstall slip ring in swivel assembly and hook into swivel.

(g) Check for normal hook operation.

Embedding the conductors in the core of the cable protects them from damage due to rough handling and from adverse environmental conditions.

Three design features have been incorporated in the single-point hoist design to provide good cable fatigue strength. Abrasion type wear, which results from cable slippage on the drum during the starting cycle, will be appreciably reduced by the use of a hard rubber jacket molded onto the aluminum drum; an added advantage is that drum wear is also reduced. In addition, a drum diameter greater than the minimum permissible based on a standard wire diameter to drum ratio is used. This results in a stress level in the cable, due to winding on the drum, which is 42 pct lower than that which would result if the minimum permissible diameter of 24 inches were used. By the use of a single layer design on the drum, the abrasion wear which would occur at cable crossovers if a multiple layer design were used is completely eliminated.

The estimated stretch in the cable, based on preliminary tests conducted on a 5-foot sample length of 18 x 19 cable, is equal to 1.53×10^{-6} inches per foot of cable extended per pound of load or at maximum load and cable length:

$$\text{Cable stretch} = (1.53 \times 10^{-6})(100)(40{,}000) = 6.22 \text{ inches} \qquad (26)$$

MISCELLANEOUS COMPONENTS

Hook-Swivel Assembly

The hook-swivel assembly provides the means for attaching cargo to the single-point hoist cable. It permits free rotation of loads about the cable centerline and transmits electrical control and indicator signals from the hoist cable to the hook.

The assembly consists of a swivel assembly and a hook assembly. These assemblies are integral units and can be quickly separated for either maintenance or replacement.

Swivel

The swivel assembly consists of a main housing assembly and a slip ring assembly. The main housing is threaded to accept the cable and contains a sealed grease packed thrust bearing. Two lugs permit attachment to the hook assembly. The slip ring assembly, which is a completely sealed unit, is bolted into the lower part of the housing. An electrical conduit, with suitable shielding to prevent wear due to rough usage, connects the slip ring to the hook assembly. An AN 4064 type dehydrator, suitably protected from abuse, is also installed to indicate excessive amounts of moisture contamination without the need for disassembly. The complete assembly weighs approximately 48 pounds.

Hook

The hook assembly is of the open throat or self-loading design. A toggle linkage is used to lock the load beam in position. A rotary solenoid provides power to open the locking linkage which allows the load to open the load beam. The load beam is restrained, following load release, by a replaceable rubber bumper. The beam is returned to the locked position by an integral spring. Integral microswitches are incorporated to provide the signal required to indicate load beam position. A compression spring mounted at the load beam pivot point and a microswitch provide the means to initiate the automatic touchdown release mode. While the load is being supported on the load beam, the spring is compressed and the microswitch is not actuated. When the load on the beam is reduced below 150 pounds (by putting the load on the ground), the spring extends and the microswitch is actuated. This signal causes the solenoid to open the locking linkage and thereby release the load.

The solenoid is completely enclosed and sealed in a weathertight housing.

All microswitches are completely sealed. An AN 4064 type dehydrator, suitably protected from abuse, is installed to indicate excessive amounts of contamination without the need for disassembly. A manual release control knob is mounted on the outside of the hook. It permits manual opening of the same locking linkage as the solenoid and provides a redundant hook release method. A knob design is used instead of a lever, since it is less subject to accidental fouling from slings and other load attachment type equipment.

Two handles, one on each side, are provided to facilitate ground handling of the hook. A rugged keeper is spring loaded to permit load rings to be slid onto the load beam with ease and yet to prevent them from sliding off under adverse loading conditions. The complete hook assembly weighs 102 pounds.

Decoupler

The single-point hoist decoupler (or isolator) is a nonlinear spring for which the spring constant increases as the load increases. By varying the stiffness of the spring with load, it is possible to maintain a constant natural frequency for the load and cable system. This system frequency is sufficiently removed from the one per revolution of mean ratio excitation frequency to eliminate any tendencies of vertical oscillation (vertical bounce).

A liquid spring is used in this application. It is provided with a 10,000-pound preload and low friction seals to minimize the operational coulomb friction (less than 5 pct of the applied load) and breakaway force (less than 250 pounds).

The incorporation of a reentrant short pipe type orifice in the liquid spring retards the return stroke of the isolator when a load is air dropped. This eliminates the use of shock struts and causes a consequent reduction in both weight and complexity.

An integral load cell permits cable loads to be measured. A charging cylinder, pressurized by the aircraft's utility hydraulic system, compensates both for temperature induced pressure changes and any leakage that may occur.

CONTROLS AND INDICATOR SYSTEM

Single-point hoist and hook controls are available to both pilots and to the hoist operator (aft-facing pilot).

The master control for the hoist is located on the console between the pilots. It consists of a master switch, which energizes either the single- or four-point hoist system, and a station selector switch, which permits hoist operation by either pilot, copilot, hoist operator, or all three. A three-position rocker switch located on all three collective pitch sticks allows the hoist to be raised, lowered, or stopped. This switch is spring

loaded to the off position. A guarded cable shear switch located in the overhead console between the pilots permits the cable to be cut in the event of an emergency.

The free reel release lever located in the center console on the floor between the pilots permits the cable and hook assembly to be jettisoned in the event of an emergency. The master switch ensures that only the appropriate hoist system will be capable of free reeling.

A cable cutter test panel is located in close proximity to the shear switch. It can be used to permit preflight checking of the firing circuit.

A similar shear switch and a test panel are located on the bulkhead to the right of the hoist operator. A free reel release lever is also provided.

Cable length and cable load indicators are provided for both the pilot and hoist operator.

The master control for the cargo hook is located in the center console between the pilots. It contains a station selector switch which permits hook operation by either pilot, copilot, or hoist operator, and a mode selector switch which can be placed on ELEC. REL., AUTO. REL., or SAFE. Push-button switches on all three cyclic control sticks permit the hooks to be opened.

Lights in the main advisory panel and at the hoist operator's station indicate when the hook is in the AUTO REL. condition or when it is in the OPEN position.

FOUR-POINT HOIST SYSTEM

HOIST

The four-point hoists (Figure 73, page 261) are universally mounted and are of the one-part, single-reeved type. The universal mounting, as shown by Figure 69, page 253, permits the hoist to be pivoted through a cone with an included angle of 60°. This enables attachment to loads of the wide variety of physical dimensions that are within the load carrying capability of the H.L.H. Figure 3, page 13, shows the space envelope for the load attachment points that are within the hoists capability without exceeding the 60° included cone angle.

Since the hoist is mounted in a universal joint, small side loads will be reacted in the level wind support structure. Figure 44 illustrates this condition. To reduce wear of both cable and bellmouth, two nonpowered, polyurethane rubber coated rollers are integrated into the bellmouth assembly.

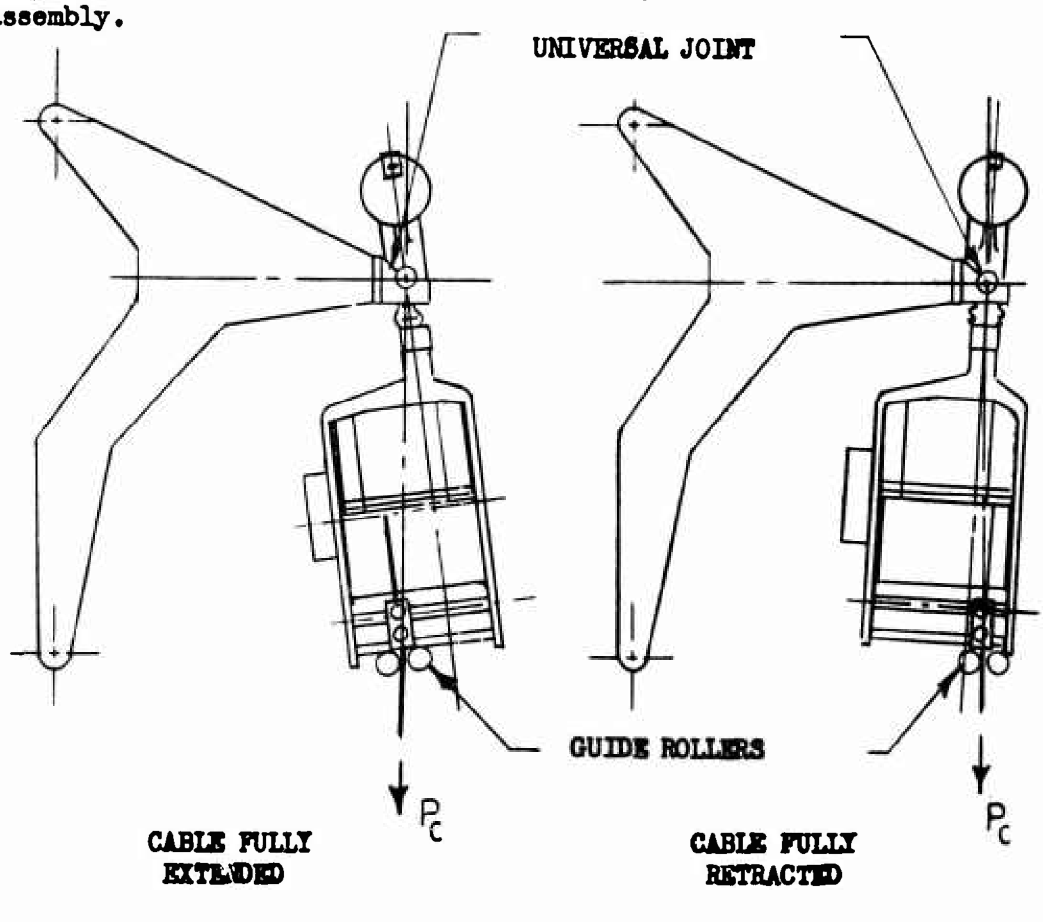


Figure 44. Four-Point Hoist Attitude at Cable Extremes.

The cable is wrapped in a single layer on a grooved drum which is covered by a 0.25-inch-thick molded rubber jacket which is attached to the drum by a pin and clamp fitting. A level wind mechanism, consisting of a ball screw driven roller bellmouth assembly, ensures even distribution of the cable across the drum. Scrub rollers are used to provide the tension required to pull the cable off the drum and are adjustable to compensate for wear.

The power required to drive the scrub roller and the ball screw is supplied by a chain drive from a drum mounted sprocket. The roller bellmouth assembly is supported on a ball screw nut and a reaction pipe. The reaction pipe provides the restraint necessary to transmit the rotary motion of the ball screw into a linear motion of the roller bellmouth-nut assembly. The cable cutters, which are used to shear the cable in the event of an emergency, are located in the roller bellmouth assembly.

A Weston brake is provided to prevent the load from dropping in the event of a power failure. It functions automatically and is located between the first and second stages of the main drive gear train.

A hydraulically actuated free reel clutch similar to that used in the single-point hoist (see page 136) is incorporated in the hoist.

A completely sealed slip ring assembly is provided to transfer electrical signals from the aircraft to the main suspension cable on the hoist. A quick disconnect fitting permits hoist cable removal without affecting the slip ring assembly.

Microswitches are provided to limit cable travel, and a cable length potentiometer is provided. A feedback control system potentiometer and clutch assembly is also integrated in the hoist drive train.

An anti-backlash cover prevents the cable from jumping off the drum in the event that a load is air dropped.

The power gear train consists of four reduction stages with an overall reduction ratio of 514.4 to 1. The first and second reduction stages, 4.52 to 1 and 6.27 to 1, are conventional spur gears. The third and fourth stages, 4.90 to 1 and 3.70 to 1, are conventional planetary gear sets. All of this gearing is splash lubricated and completely sealed. Oil fill, drain, and level plugs are provided.

HYDRAULIC SYSTEM

Pump

The hoist pump is a yoke type in which the inclination of the yoke establishes the angle of the swashplate, to which the displacement of the pump is proportional. When the yoke is rotated across the center, or no-flow, position the direction of flow through the pump can be reversed. The pump displacement is controlled by a single stage servo positioner. The signal

to the valve is electrical, and control power is supplied by the replenishment hydraulic system. If either the control signal or control pressure is interrupted, the pump returns to the no-flow position. The magnitude and polarity of the electrical control signal determines quantity and direction of pump flow. The control system is designed to permit seven delivery positions: 1/4 flow, 1/2 flow, and full flow in both directions and zero flow. The pump displacement is 1.8 cu in./rev. It is rated at a flow of 45 gpm at 6000 rpm and weighs 20 pounds.

Motors

The motors are of the fixed displacement type whose direction of rotation depends on the direction of flow. They have a displacement of 0.95 cu in./rev and an output power, at 3000 psi, of 17.5 horsepower. They weigh 10 pounds each.

Hydroelectrical Feedback System

The basic hydraulic system is a pressure demand type in which the direction and rate of flow are established by control of the hoist pump and the pressure developed is determined by line losses and the load. This type of system eliminates the heat generation of a system which uses a pressure compensated pump at anything less than full load. The system is closed and does not contain a reservoir. Losses from the system through pump and motor leakage are made up by a separate replenishment system. The pump supplying the replenishment system also provides control pressure to the hoist pump controller and cooling flow to the hoist pump.

Synchronized operation of all four hoists is obtained by use of a hydroelectrical feedback system which utilizes servo controlled flow divider valves to control the flow to individual hoist motors, as shown by Figure 45, page 146.

The division of flow between forward and aft and between port and starboard hoists is established by servo controlled flow divider valves. The signal to the flow dividers is derived from a comparison of the signals from rotary potentiometers on each hoist. The output signals from the potentiometers on the forward pair of hoists supply two amplifiers. The output of one amplifier is proportional to the algebraic sum of the signals, the output of the other is proportional to the algebraic difference between the signals. The potentiometers on the aft pair of hoists supply two similar amplifiers. The outputs from the summing amplifiers of both forward and aft pairs are supplied to another amplifier, the output of which is proportional to the difference between the two supply signals. The output of this amplifier controls the flow divider valve which determines the distribution of flow between the forward and aft hoist pairs.

Division of a flow between port and starboard hoists, forward or aft, is effected by a flow divider valve which is controlled by the signal from the differencing amplifier of the appropriate hoist pair.

During lowering of a load the signal to the flow divider valve to correct

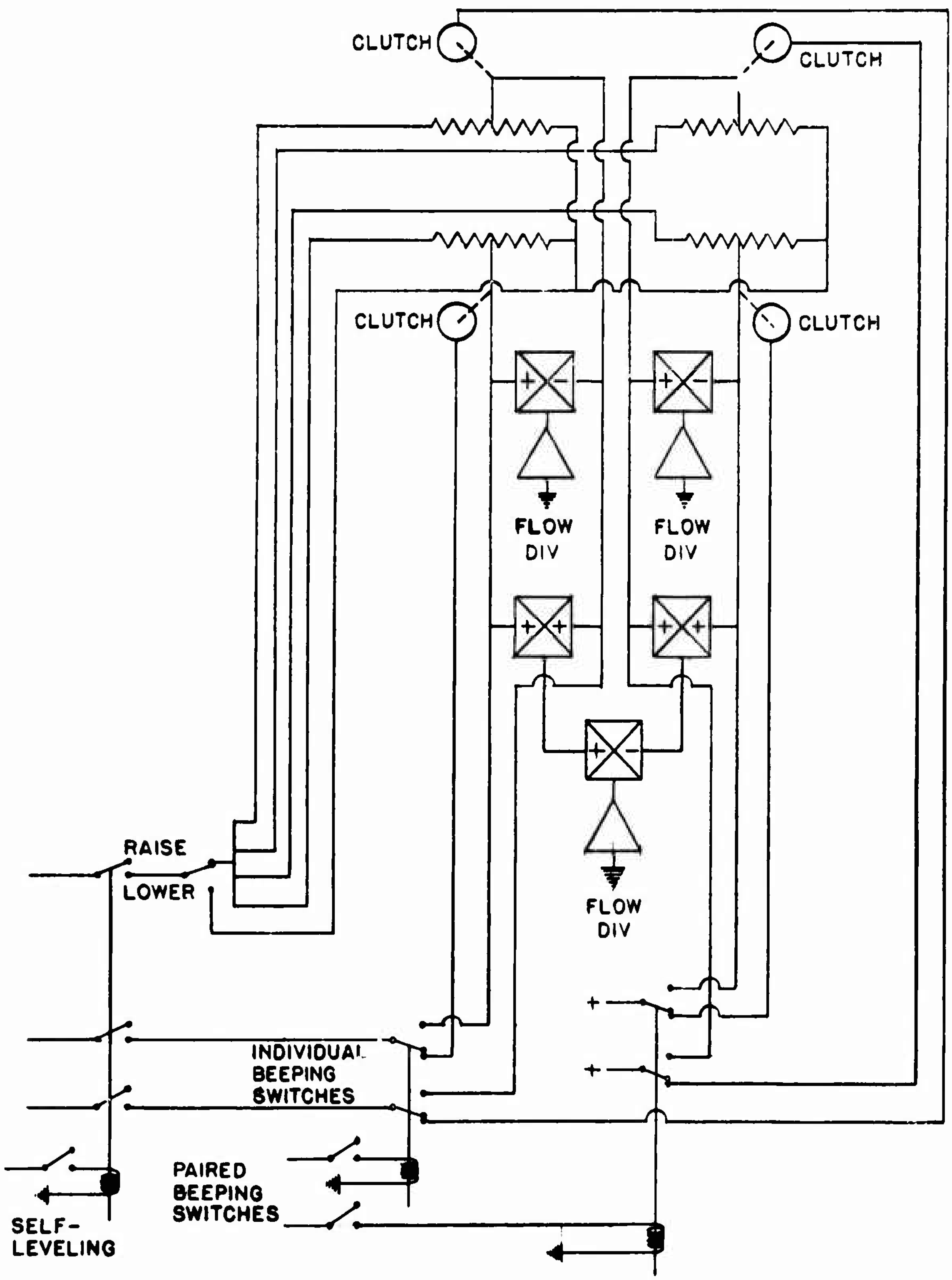


Figure 45. Hydroelectrical Feedback System, Four-Point Hoist.

an error must be of the opposite sign to that to correct an error when raising a load. To achieve this, the polarity of the potentiometers is reversed when the direction of movement is reversed.

Operation of the hoists individually is accomplished by supplying bias signals to the two relevant flow divider valves to direct flow to one hoist only. During this beeping operation, the output of the pump will be 1/4 of its full flow delivery to prevent overspeeding of the hoist motor.

The forward or aft hoists may be operated in pairs by supplying a bias signal to the forward/aft division flow divider and a centering signal to the other appropriate flow divider valve. During this dual-beeping operation, the output of the pump will be 1/2 full flow delivery to prevent overspeeding of the hoist motor.

Magnetic clutches between the potentiometers and the hoists will disengage during beeping and reengage for collective operation. Thus the system can be trimmed for any load shape and the established cable lengths will be maintained during collective operation.

The system will be somewhat load sensitive. A heavily loaded hoist will lag initially until the bias on the flow divider valve is sufficient to compensate for the unequal loads. The hoists will then operate at the same nominal rate. If collective operation is stopped at this stage and errors are recovered by beeping, the hoists will continue to operate at the same rate and the load will remain level when collective operation is resumed. This occurs because, during beeping, the potentiometer settings and hence the signals to the flow dividers were unchanged and the resistances of the circuits remained balanced.

System Components

The flow divider valves provide the means of varying the resistance of each hydraulic circuit to achieve equal flow rates to each hoist regardless of the load. The valve is comprised of two stages. The first stage is a torque motor operated, flapper-nozzle type which establishes pilot pressures in response to the differential signals in the coils. The second stage is two spool type throttle valves which are positioned by pilot pressures from the first stage. In operation, an electrical signal to the valve causes the second stage spool to move from the center position and provide increased restriction to flow in one leg. The increase in restriction is proportional to the magnitude of the electrical signal.

Two sizes of valves are required, one handling a total flow of 45 gpm to control the division of flow between forward and aft hoist pairs, the other to control the division of flow between port and starboard hoists and to be capable of passing 22.5 gpm total flow. Two of the latter type are to be used. The three valves will be mounted together to make one compact unit and to eliminate connecting plumbing.

The potentiometers are high precision 10 turn type with a total resistance error of $\pm$ 1/2 pct, a linearity error of .03 pct, and a resolution error of .03 pct.

Electromagnetic clutches are used to drive the potentiometers. Since they are of the magnetic particle type, they will not be affected by temperature and humidity. Wear and adjustment requirements will be negligible, since there is no contact between the driver and driven elements.

The main system lines will be of stainless steel with a 1-inch O.D. and a .065-inch wall thickness. Lines to the individual hoists will also be of stainless steel with a 3/4-inch O.D. and a .049-inch wall thickness.

ERROR ANALYSIS

A hydroelectrical feedback system (page 145) is used to provide synchronous operation of the four-point hoist system to minimize the variation in hoist operating speeds due to differences in load, motor leakage, mechanical and hydraulic efficiencies, and errors introduced by the flow control components. Errors are introduced by the sensing elements, by hysteresis in the control components, and by drum and cable diameter tolerance.

Difference in hoist positions can be caused by the errors in the control elements (steady state errors) or by bias in the system necessary to accommodate different loads, leakages, or efficiencies (dynamic errors). The latter can be removed by beeping the offending hoist (or hoists) until the load is level (or at the desired attitude) and then continuing collective operation. As this dynamic error can be eliminated, it will not be considered in the error analysis. The steady state error is that difference in hoist position which cannot be sensed by the potentiometers (resolution error), or is insufficient to actuate the flow divider valve, and that difference in hoist position which is necessary to compensate to linearity and total resistance tolerances of the potentiometers. The specification of the potentiometers will permit a maximum total resistance error of $\pm$ 1/2 pct, a maximum linearity error of $\pm$ .03 pct, and a maximum resolution error of $\pm$ .03 pct. The flow divider valve will have a hysteresis of not greater than 1 pct of its rated signal, i.e., the signal to direct all flow to one port, and the control system is so designed that a discrepancy of 6 inches in hoist position will produce this rated signal.

The steady state error is composed of two components: one part is dependent upon the amount of cable travel; the other is independent of cable movement. The total resistance tolerance produces an error dependent upon cable travel; other control system errors are independent of cable movement. In assessing the steady state error, it is unnecessary to consider the linearity of the flow divider valves or amplifiers, as in the steady state these components will always be operating at the same point of their characteristic. The errors made possible by the linearity and resolution tolerance of the potentiometers is a function of the total hoisting capacity;i.e., the error introduced by the linearity error of .03 pct is .18 inch.

The tolerance on total resistance produces an error of .5 pct of the movement of the hoisting (i.e., after L feet of hoisting the error could be .005 L feet).

Total errors in the system are summarized below:

	Percent Errors	Error in a 50-Foot System (inches)
Errors Dependent on Cable Travel:		
Cable drum diameter tolerance	±.045	±.006L
Cable diameter tolerance	±.020	±.003L
Pot. total resistance tolerance	±.50	±.060L
Errors Independent on Cable Travel:		
Pot. linearity tolerance	±.03	±.18
Pot. resolution	±.03	±.18
Flow divider valve dead band	-	±.06

where: L is the distance traveled in feet

With the data shown above, it is possible to derive a formula describing maximum cable length discrepancy between any two hoists:

$$\text{Error, inches} = .138 \times \text{cable travel in feet} + 0.84 \quad (27)$$

These data are shown graphically in Figure 46.

The total system error may be effectively reduced in service by the use of a calibrated, parallel connected, trimming potentiometer on each hoist. A one-time check-out procedure, which requires extending the cables a full 50 feet and then reeling in to the in-limit stops, is required. The difference in cable lengths extended after the "fastest" hoist reaches its limit stop is measured and the trimming potentiometer is adjusted to compensate for the error of the out-of-time hoist(s). This adjustment may compensate for errors due to potentiometer total resistance tolerance and could theoretically reduce the 7-3/4-inch error in 50 feet to approximately 3 inches. Practical considerations, such as the desire to reel in at no load, would probably reduce the actual error to a value slightly greater than 4 inches in 50 feet. This approach will require further investigation.

CABLE

The four-point hoist cables are stainless steel and are of 18 x 19 non-rotating construction. They have an outside diameter of 0.79 inch and a guaranteed minimum breaking strength of 49,000 pounds. The individual wires are .035 inch in diameter and are made from heat treated type 302

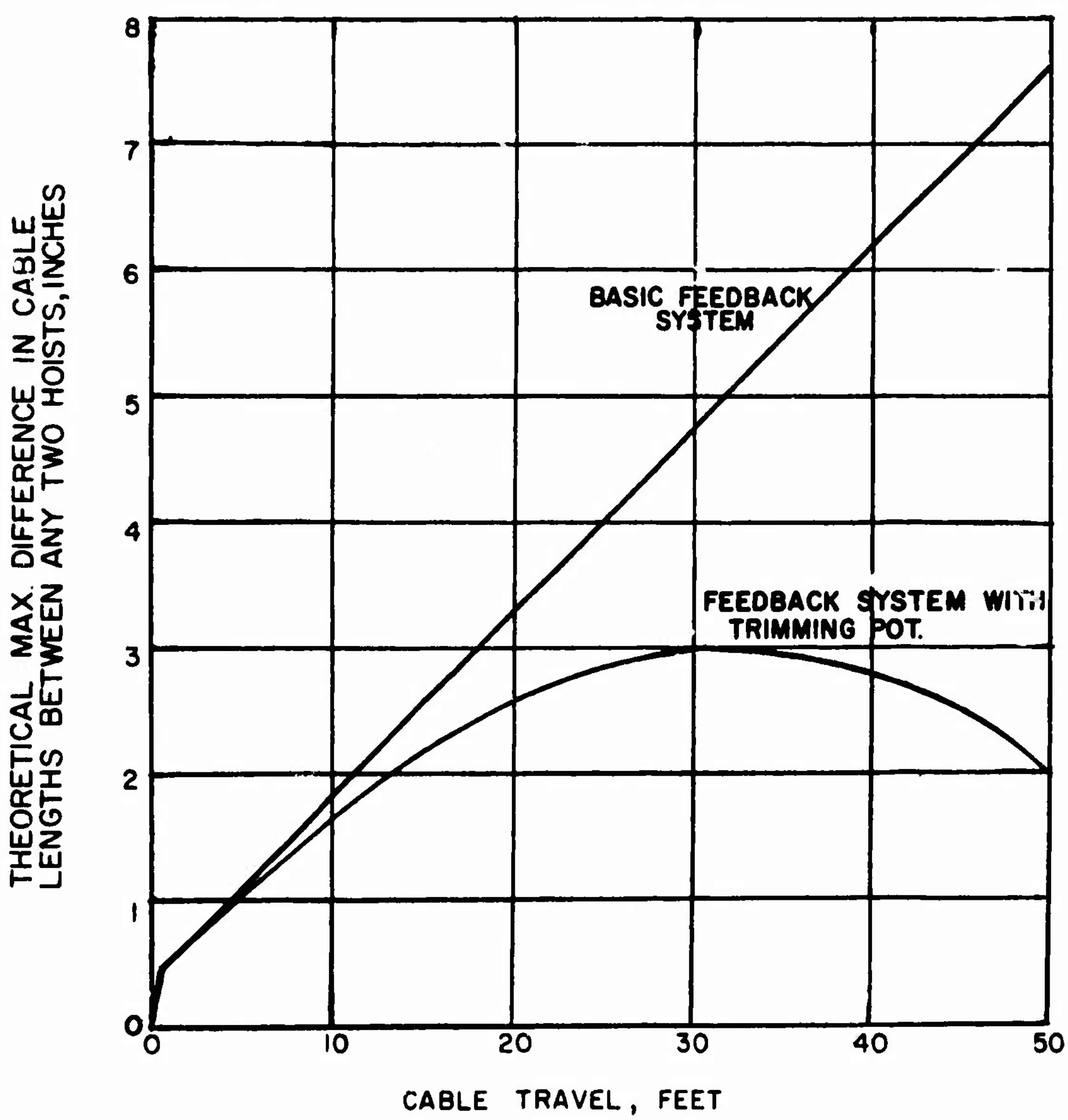


Figure 46. Difference in Cable Length vs Cable Travel, Four-Point Hoist System.

stainless steel. The 18 x 19 construction, 18 strands with 19 wires per strand, gives a flexibility greater than that attained in the 7/8 diameter 18 x 7 cable presently used in the CH-54A main cargo hoist.

The core of the cable contains seven electrical conductors of stranded wire construction. They are helically wound and encased in a tough, resilient, plastic jacket. Five of these conductors are required to transmit the electrical power required to operate the actuating solenoid in the cargo hook. Two conductors are spares; thus field level maintenance personnel can "wire out" defective conductors without removing the cable. This "wiring out" procedure is identical with that described for the single-point hoist cable on page 139. Embedding the conductors in the core of the cable protects them from damage due to rough handling and from adverse environmental conditions.

Three design features have been incorporated in the four-point hoist design to provide good cable fatigue strength. Abrasion type wear, which results from cable slippage on the drum during the starting cycle, will be appreciably reduced by the use of a hard rubber jacket molded onto the aluminum drum; an added advantage is that drum wear is also reduced. In addition, a drum diameter greater than the minimum permissible based on a standard wire diameter to drum ratio is used. This results in a stress level in the cable due to winding on the drum, which is 58 pct lower than that which would result if the minimum permissible drum diameter of 14 inches were used. By the use of a single layer design on the drum, the abrasion wear which would occur at cable crossovers if a multiple layer design were used is completely eliminated.

The estimated stretch in the cable, based on preliminary tests conducted on a sample length of 18 x 19 cable, is equal to 4.75×10^{-6} inches per foot of cable extended per pound of load or at maximum load and cable length:

$$\text{Cable stretch} = (4.75 \times 10^{-6})\ (50)\ (11{,}550) = 2.75 \text{ inches} \qquad (28)$$

MISCELLANEOUS COMPONENTS

Hook-Swivel Assembly

The hook-swivel assembly provides the means for attaching cargo to the hoist cable. It permits free rotation of loads about the cable centerline and transmits electrical control and indicator signals from the hoist cable to the hook. The use of a swivel assembly permits individual loads to be carried on the four-point hoists. This feature will permit individual loads, such as fuel bags, to be carried to and off loaded at separate sites. The assembly consists of a swivel assembly and a hook assembly. These assemblies are integral units and can be quickly separated for either maintenance or replacement. Figure 74, page 263, shows the hook-swivel assembly. The relative sizes of both the 40,000 pound and the 11,550 pound capacity hooks are also shown.

Swivel

The swivel assembly consists of a main housing assembly and a slip ring assembly. The main housing is threaded to accept the cable end fitting and contains the sealed, grease packed thrust bearing. Two lugs permit attachment to the hook assembly. A slip ring assembly, which is a completely sealed unit, is bolted to the lower part of the housing. The complete assembly is identical in design to the single-point hoist swivel assembly described in detail on page 140. The swivel assembly weighs 21 pounds.

Hook

The hook assembly is of the open throat, or self loading, design. It is similar in design to the 40,000-pound-capacity hook used for the single-point hoist which is described in detail on page 140. One feature, the automatic touchdown release, has been eliminated in the 11,550-pound-capacity hook as a safety feature. Appendix III describes a typical four-point mission and serves to illustrate why the elimination of the automatic touchdown release is considered a safety feature. The complete hook assembly weighs 31 pounds.

Isolator

The four-point hoist isolator (or decoupler) is a nonlinear spring for which the spring constant increases as the applied load increases. The isolator consists of two air-oil accumulators, a servo valve, a housing containing two pistons, and a load cell. Any relative motion of small amplitude between the load and the aircraft is absorbed by the two accumulators. The accumulators receive and release hydraulic fluid into the chamber below the lower piston as it moves up and down relative to the housing. The resulting spring rate, therefore, is a function of the pneumatic characteristics of the accumulators. Sudden load release is damped out by the movement of the tapered pin which is attached to the upper piston through a fixed orifice.

The load position is maintained hydraulically by the action of the servo valve and isolator feedback linkage.

The accumulators contain gages to permit check-out for proper precharging and a service valve to permit recharging as required.

A separate load cell is attached between the isolator and hoist attachment fitting, thus permitting field replacement without disassembly of the isolator. Figure 75, page 265, shows the isolator.

The complete isolator assembly weighs 38 pounds.

CONTROLS AND INDICATOR SYSTEM

Four-point hoist and hook controls are available to the pilot and copilot and to the aft-facing pilot (hoist operator). In addition, a special control box with a coil cord extension is provided for use by a dismounted loadmaster. The master control for the hoist systems is located on the console between the pilots. It consists of a master switch, which energizes either the single- or four-point system, and a station selector switch, which permits hoist operation by either pilot, copilot, aft-facing pilot (hoist operator), loadmaster, or all four.

As in the single-point hoist system, the three-position, rocker-type switch located on all three collective pitch sticks allows the hoists to be raised, lowered, or stopped. The four-point hoist selector control is located as an extension of the pilots' and the aft pilot's collective pitch stick. The control consists of seven selector buttons. These buttons are marked to permit operation of hoists number 1, 2, 3, and 4; numbers 1 and 3 (forward); numbers 2 and 4 (aft); or all four hoists simultaneously. The buttons are located in precise visual orientation with the hoists to be operated, as shown in Figure 47, page 155, which is the view of the aft-facing pilot when operating the hoists.

The cable length indicators are of the tape type and are spatially oriented with the appropriate hoist.* Cable load indicators are of the dial type, to avoid confusion with the length indicators, and are also located so that they are oriented, by the eye, with the hoist whose load they indicate.

The master control for the four-point hooks is located in the center console between the pilots. It contains a station selector switch and a hook selector switch. The hook selector switch allows hook numbers 1, 2, 3, and 4 or all hooks to be operated. The actuation of the hook(s) is controlled by push-button switches on the cyclic sticks.

Lights in the pilot's advisory panel indicate when the hook load beam is open. The aft pilot is provided with similar indicator lights.

The free reel release lever (described on page 142) permits all four cable and hook assemblies to be jettisoned in the event of an emergency. A guarded cable shear actuation switch is located in the overhead console between the pilots and permits all four cables to be cut simultaneously in the event of an emergency. A hoist selector switch spring loaded to the ALL position is included in the same panel. It also permits selection of any of the individual cables for shearing in event of an emergency when carrying individual hoist loads. A cable cutter test panel is also provided to permit preflight checking of the firing circuit. Shear actuation controls are located on the bulkhead to the right of the aft-facing pilot.

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*This type of instrument correlates more naturally with cable length than the dial type.

The dismounted loadmaster's control, shown in Figure 47, is a hand held control for both hoist and hook actuation. It consists of seven hoist control selecter push buttons and is similar to that available to the pilot and copilot. A toggle switch permits the desired hoist(s) to be raised, lowered, or shut off. Four toggle switches, spring loaded to the OFF position, permit any of the four hooks to be opened. If all four are to be released at once, the "gang bar" is used to ensure simultaneous release.

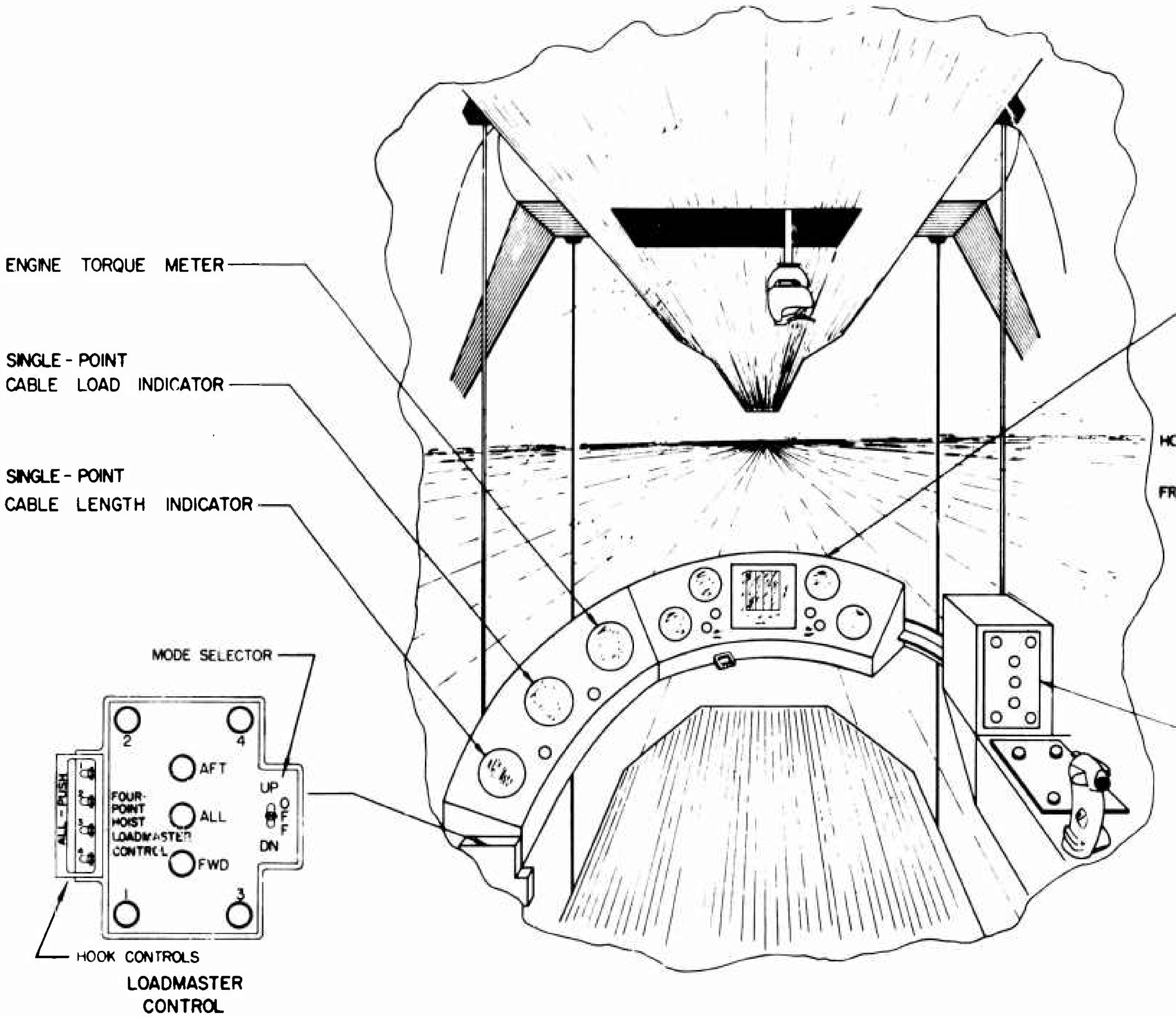
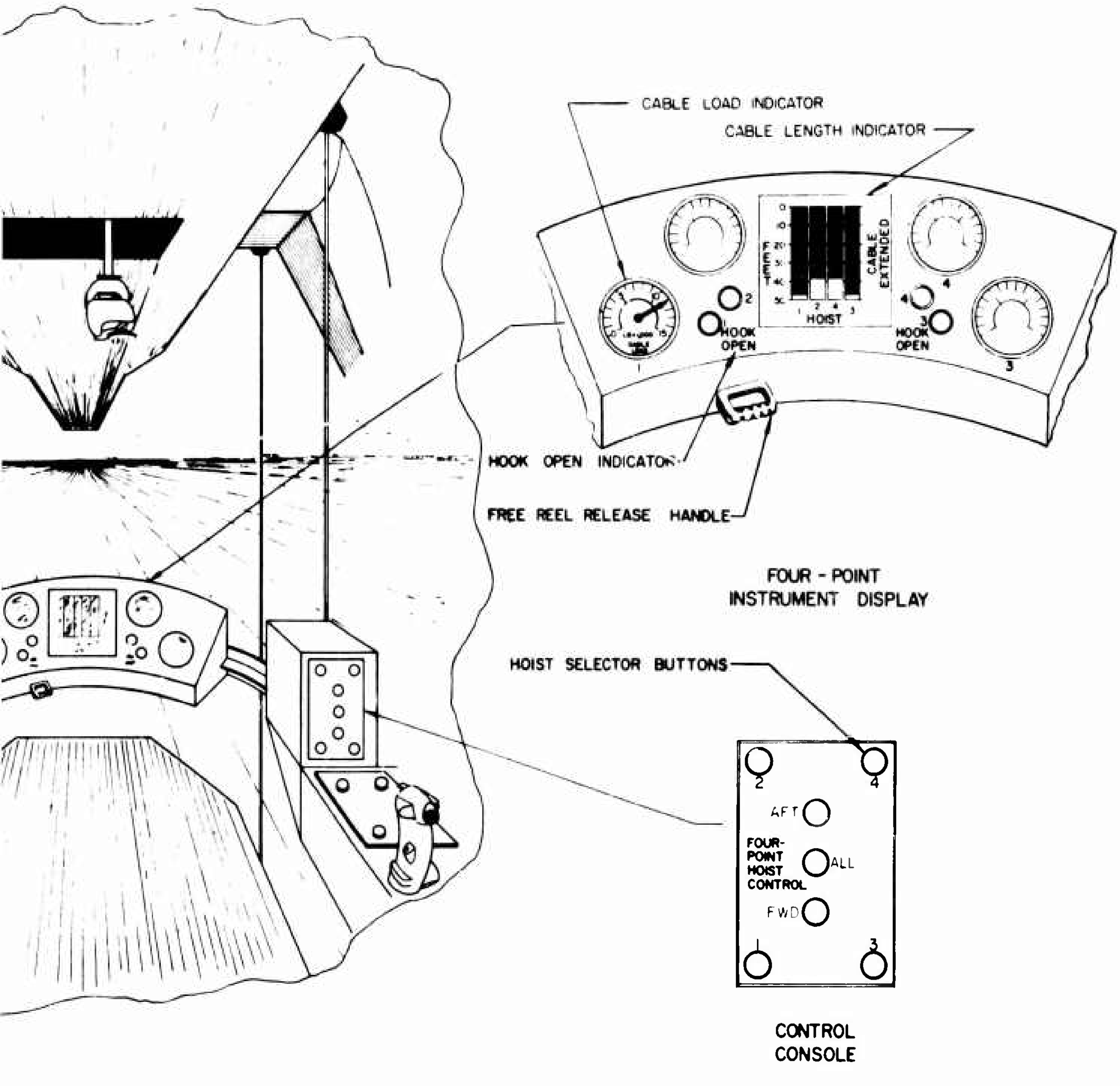


Figure 47. Controls and Indicators, Single- and Four-Point Hoists.

A.



-Point Hoists.

B,

LOAD AND STRESS ANALYSIS

INTRODUCTION

The hoist components evaluated herein have been designed to achieve a minimum service interval of 3600 cycles or 200 hours of continuous operation. One of the design objectives as set forth in the contract was that the system shall have a minimum service interval of 1200 cycles. Defining a cycle as reeling in and reeling out all the available cable at full load, the minimum service interval of 1200 cycles is equivalent to 66.7 hours of operation for both the single- and four-point hoists. The hoist bearings are selected on the basis of a minimum B-10 life of 200 hours or static nonbrinell capacity, whichever is more critical. For the most part, bearings on the cable load side of the Weston brake are selected on the basis of static capacity, while those on the input side are selected on the minimum B-10 life requirement.

The 200-hour service interval corresponds to 3600 cycles of operation as defined above. In actual operation, the load and cable travel will often be less than the maximum. The true service life of the hoist components should therefore exceed 3600 actual service cycles, and 200 hours should be used as the governing factor for overhaul.

The selection of materials for the hoist components evaluated in this study is based on Sikorsky Aircraft's extensive test and production aircraft experience. All materials considered for the hoist are currently used in similar applications on production aircraft.

DESIGN CRITERIA

Gearing

Material AMS 6260 (SAE 9310) Steel

Spiral Bevel Gears

Bending Stress, F_b = 30,000 psi

Compressive Stress, F_c = 200,000 psi

Spur Gears

Bending Stress, F_b = 36,000 psi (one-way bending)

Bending Stress, F_b = 32,000 psi (reversed bending)

Compressive Stress, F_c = 150,000 psi

Shafting

Material AMS 6260 (SAE 9310) Steel
F_{tu} = 136,000 psi

AMS 5000 (SAE 4340) Steel
F_{tu} = 200,000 psi

Bending Stress = 20,000 psi

Torsional Stress = 30,000 psi

Drums

erial 7075-T6 Aluminum

Compressive Stress, F_{cu} = 72,000 psi (ultimate load)

Housings

Material AZ 91C Magnesium Casting
ZK 60A Magnesium Forging
7075 Aluminum Forging

Planetary Carrier Plates

Material 6AL - 4V Titanium

Bending Stress, F_b = 40,000 psi

Slope, ϕ = .0010 inch per inch

Bearings

All bearings used in the single and multi-point hoist components are analyzed by the methods of References 1, 2, and 13. The bearings are designed for a B-10 life of 200 hours or for static capacity, whichever is worse. Static capacity is checked with the cable at limit load. Under these conditions, the allowable load ratings of the bearings are as follows:

Rotating bearing, $P_{allow} = 1.25\ Co$ (29)

Nonrotating bearing, $P_{allow} = 3\ Co$ (30)

where

Co = Basic static load capacity of bearing

Level wind drive bearings are designed for 200-hour B-10 life using loads derived from the power necessary to drive the level wind ball screw when the cable is at 30° side angle. Static capacity is checked using limit load conditions and 30° side angle.

SINGLE-POINT HOIST

The main drive train of the single-point hoist (Figure 71, page 257) consists of three stages of gearing. The 4.190/1 ratio first stage of gearing is a spur gear set whose pinion is driven through a Thomas coupling from drive shafting connected to the accessory gearbox. Torque flows through the input pinion and the input idler to the input gear. The output of the first stage gear drives the input side of the Weston brake. The Weston brake is arranged to raise the load with all components driving as a unit, to lower the load in the reverse drive direction with the plates slipping, and to hold the load with the power off. The output of the Weston brake is splined to a quill shaft connected to the input side of the free reeling clutch. This clutch has drive plates that are preloaded by Belleville washers and contains a hydraulic cylinder that can compress the washers in the event of an emergency, thereby releasing the drive train and allowing the load and cable to strip off the drum. The output of the free reeling clutch is a quill shaft connected to the second stage 513/1 reduction ratio spur pinion and gear set.

The third reduction stage consists of a 30.111/1 ratio compound planetary with one fixed ring gear and one output ring gear. The driving sun gear of this planetary is integral with the second stage gear shaft. Since the planetary plates carry no load, they are used only to position the planetary. The main drum is driven directly by the output ring gear of the compound planetary. All other components, such as the level wind, slip rings, and cable length potentiometer are driven off the main drum through spur gears.

The normally fixed input housing is isolated from the mounting structure by bearings allowing the reaction torque to be absorbed by a liquid spring load isolator.

Design Data

Normal cable load	=	40,000 lb
Limit cable load	=	100,000 lb
Yield cable load	=	115,000 lb
Ultimate cable load	=	150,000 lb
Cable diameter	=	1.39 inches
Useful cable length	=	100 feet
Mean drum diameter	=	34 inches
Cable angle - static	=	$\pm$ 30° (starboard, port) + 30° (forward) - 60° (aft)
Cable angle-dynamic	=	$\pm$ 15° (starboard, port) + 15° (forward) - 30° (aft)
Input speed	=	3000 rpm
Overall reduction ratio	=	443.218 to 1
1st stage spur ratio	=	4.190 to 1
2nd stage spur ratio	=	3.513 to 1
3rd stage compound planetary ratio	=	30.111 to 1
Pitch line velocity	=	60.2 fpm

Figure 48 is a schematic of the basic drive gearing arrangement.

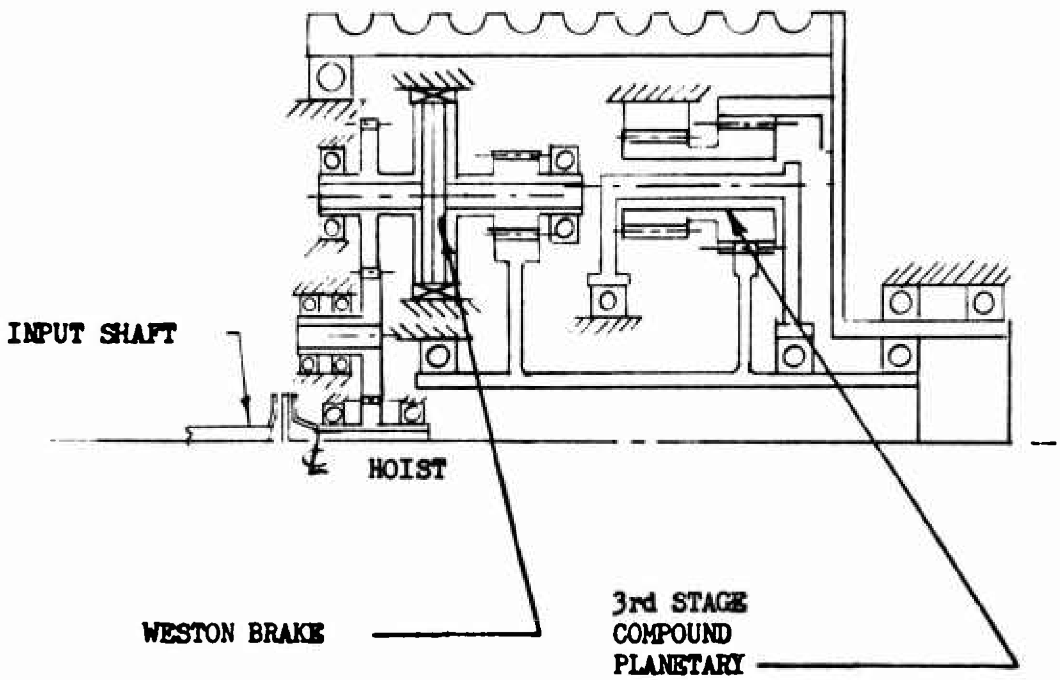


Figure 48. Gearing Schematic, Single-Point Hoist.

Drum Design

In the Phase I Design Analysis section of this report the, the single-point hoist width and diameter were conservatively chosen at 15 inches and 39 inches respectively for 100 feet of cable to maintain the single-rotor heavy lift helicopter's lateral cyclic stick movements within desirable limits, see page 18. A more detailed aerodynamic trim analysis, based on the 3-inch permissible control stick travel of Reference 10,* indicates that the allowable lateral C.G. shift is 9 inches to the right and 8.4 inches to the left. Figure 49 shows the C.G. shift due to hoist loads.

*Note: The longitudinal control displacement of 3 inches from the initial trim position is considered to be applicable to lateral trim conditions in the absence of a specific limitation for lateral stick displacement limits.

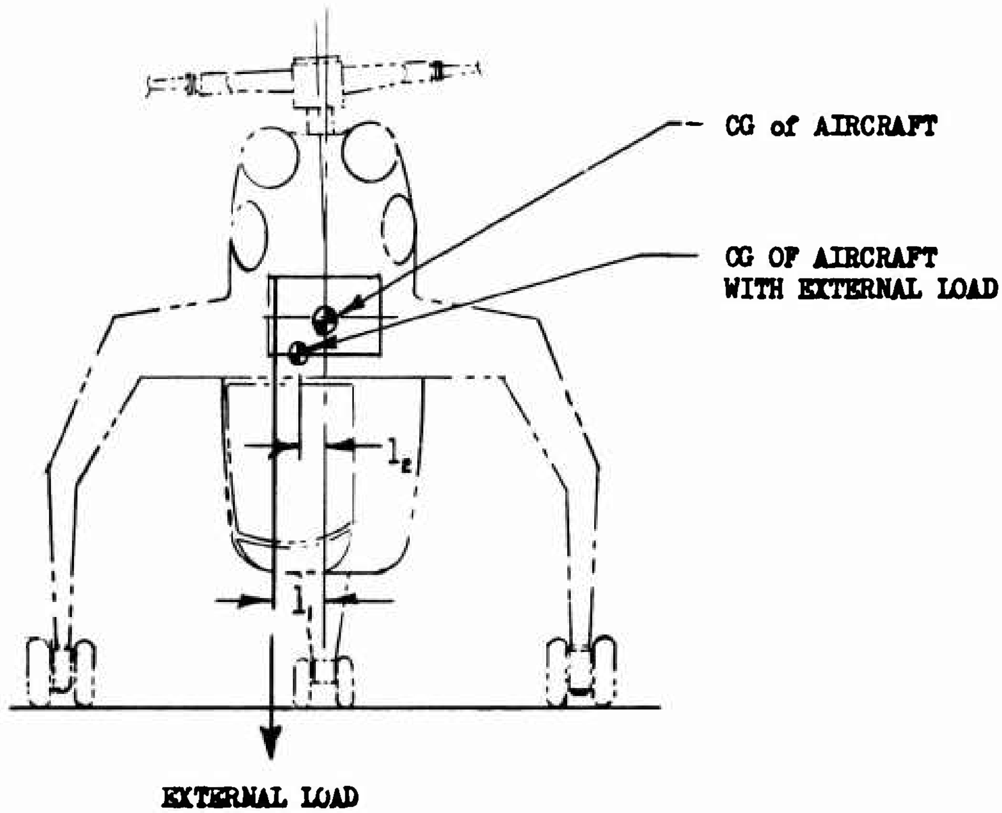


Figure 49. Center-of-Gravity Shift Due to Hoist Load At Maximum Limits of Cable.

From Figure 49,

$$\frac{l_1}{l_2} = \frac{\text{Aircraft gross weight}}{\text{External load}} \tag{31}$$

For an external load of 40,000 pounds, the single-rotor heavy lift helicopter of Reference 4 has a gross weight of 79,071 pounds. Using the minimum permissible C.G. shift equal to 8.4 inches (to the left) and solving for l_1 we obtain

$$l_1 = 8.4 \frac{79,071}{40,000}$$

$$l_1 = 16.6 \text{ inches}$$

On the basis of this analysis, a drum diameter as small as 24.0 inches can be utilized. A drum mean diameter of 34 inches has been chosen, however, to provide for good cable life and adequate space for the hoist gearing and housings.

With this size drum, 12 cable wraps are required for 100 feet of cable. With the 12 active and 3 "dead" wraps spaced at a 1.5- inch pitch, the drum width is 22.5 inches. The total cable travel is 18 inches (or 9 inches to right or left of the aircraft centerline).

Utilizing the drum analysis of page 30, the drum thickness for a 34-inch 7075 aluminum forged drum is 1.44 inches.

Major Structural Members

Figure 50 shows the location of the major structural bearings in the single-point hoist.

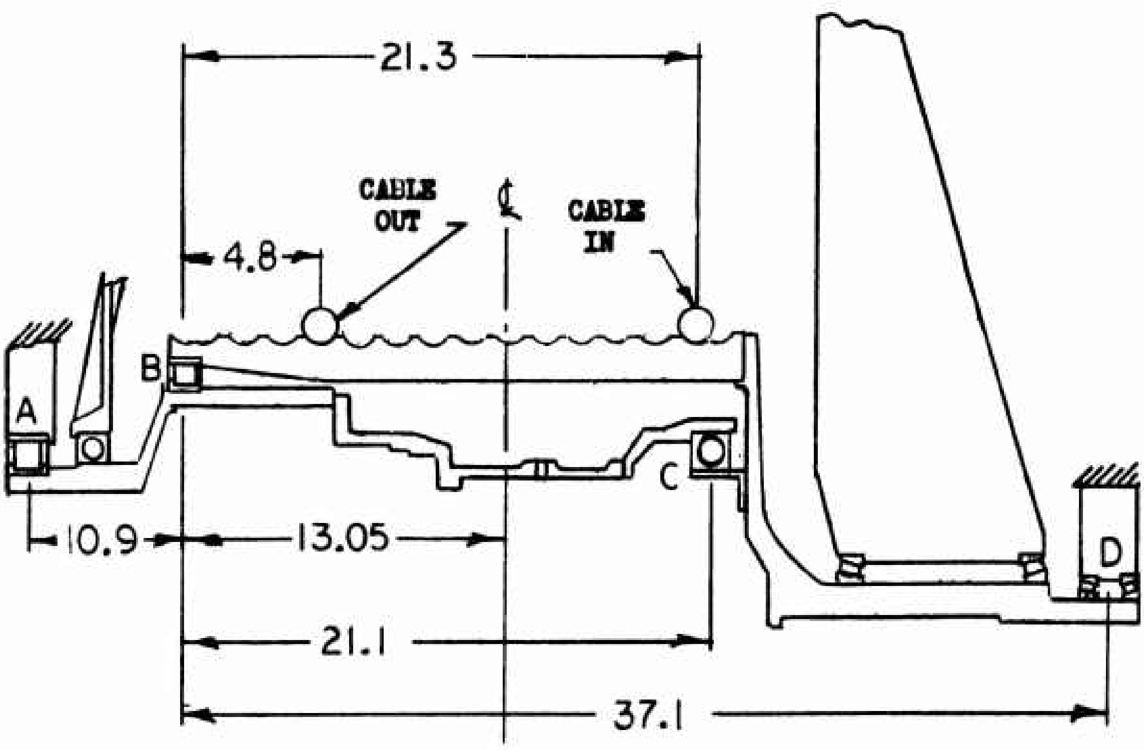


Figure 50. Drum and Support Structures, Single-Point Hoist.

Cable side loads are transmitted from the bellmouth, through the level wind ball screw, and into the level wind arm. On the main structure, the Timken bearing drum support reacts all the thrust loads from the level wind arm. The end view shown in Figure 51 locates the load isolator and fore and aft maximum cable limits.

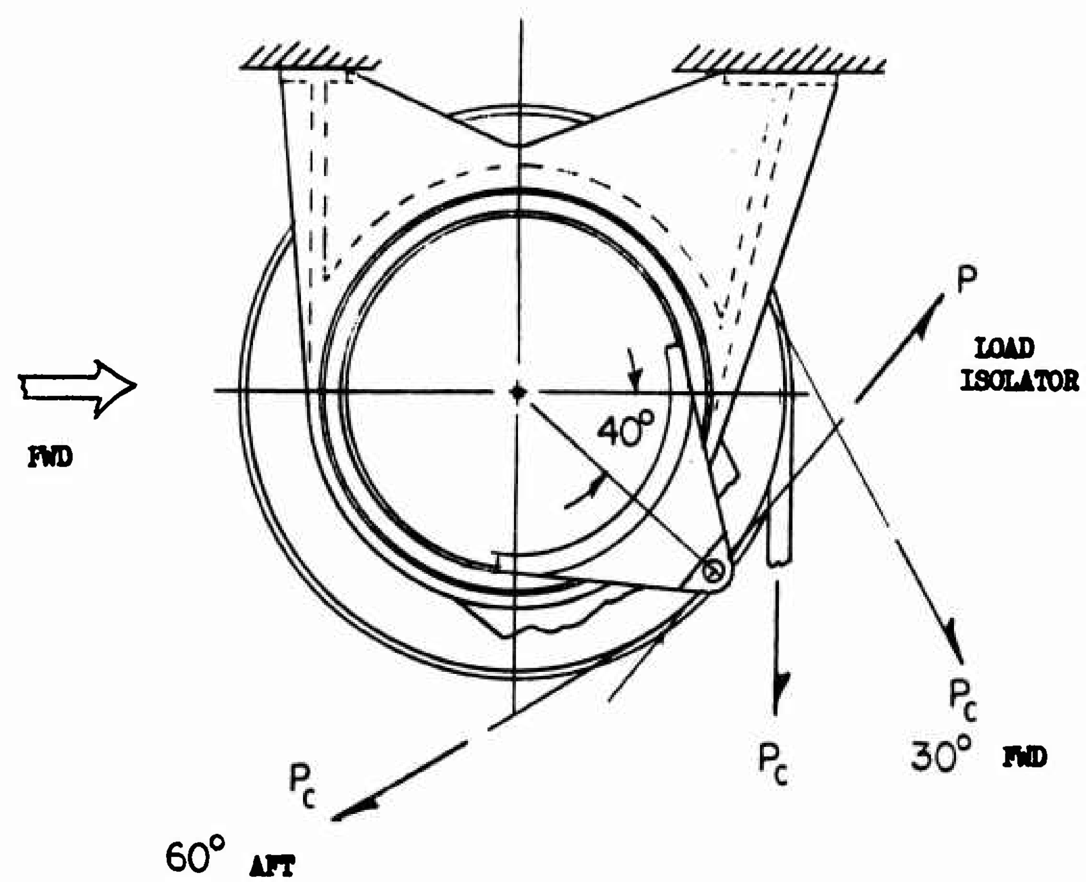


Figure 51. Load Isolator and Support Structure, Single-Point Hoist.

Table XXIII summarizes the bearing reactions, A, B, C, and D, for various load conditions.

TABLE XXIII

MAJOR SUPPORT STRUCTURE BEARING REACTIONS AT P = 150,000 POUNDS, SINGLE-POINT HOIST

Cable Position	Cable Angle	R_A Vertical	R_A Horizontal	R_B Vertical	R_B Horizontal	R_C Vertical	R_C Horizontal	R_D Vertical	R_D Horizontal
Full in	0°	72,600	99,570	80,310	4,580	38,340	10,590	108,860	6,030
Full in	30°fwd	79,230	74,820	70,300	42,010	34,890	23,450	93,510	55,870
Full in	60°aft	97,340	142,430	42,880	-60,260	25,490	-11,670	57,020	-80,290
Full out	0°	21,120	99,570	158,520	4,580	64,820	10,590	56,220	6,030
Full out	30°fwd	34,650	49,100	138,010	81,110	57,820	36,680	49,650	30,550
Full out	60°aft	71,610	187,010	81,980	-127,990	38,720	-34,600	31,700	-36,440

Figure 52 is a sketch of the input housing showing the critical section and bearing reactions for the worst case load (cable out, 30° forward).

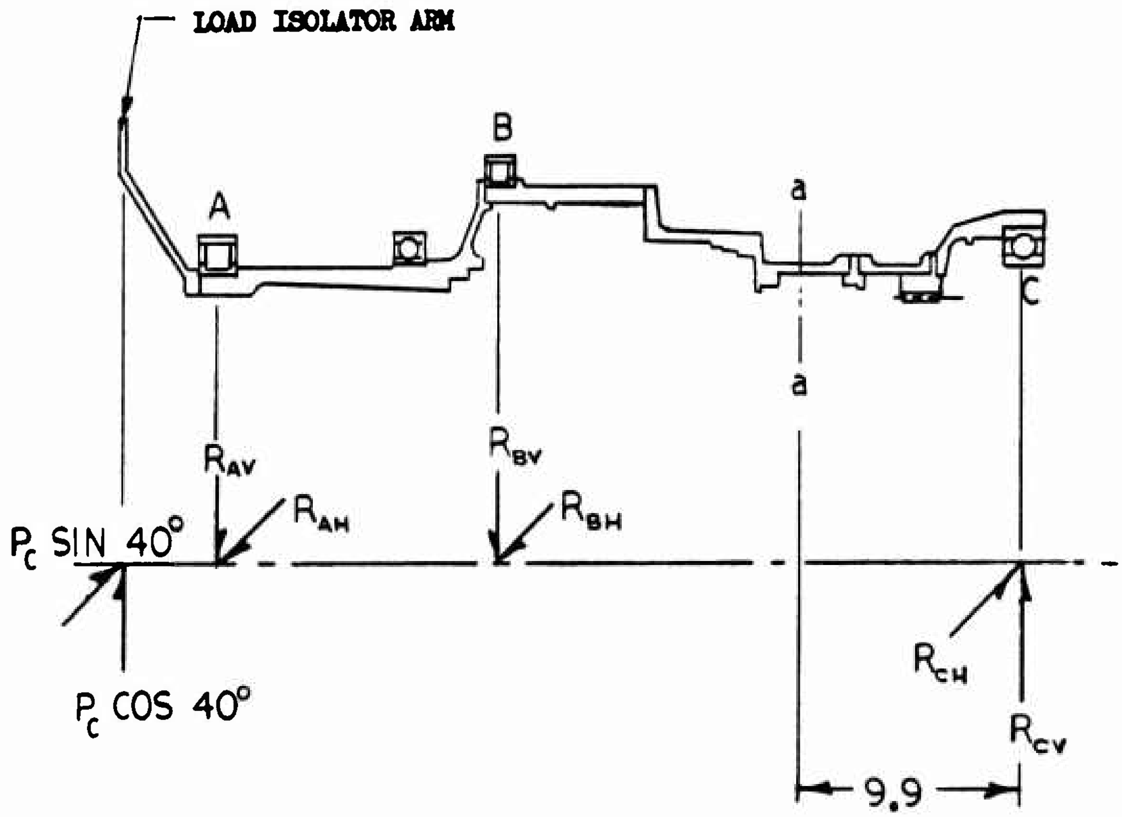


Figure 52. Input Housing, Single-Point Hoist.

At critical section a-a,

$$d_o = 23$$

$$d_i = 22.625$$

$$Z = \frac{\pi}{32} \frac{(d_o^4 - d_i^4)}{d_o}$$

$$Z = \frac{\pi}{32} \frac{(23.000^4 - 22.625^4)}{23.000}$$

$$Z = 76.02 \text{ in.}^3 \tag{32}$$

$$M_{bv} = 9.9\ R_{cv}$$
$$M_{bv} = 9.9\ (57,820) \tag{33}$$
$$M_{bv} = 572,420$$

$$M_{bH} = 9.9\ R_{cH}$$
$$M_{bH} = 9.9\ (36,680) \tag{34}$$
$$M_{bH} = 363,130$$

$$M_b = M_{bv} \mathrel{+\!\!\!\!\rightarrow} M_{bH}$$
$$M_b = 572,420 \mathrel{+\!\!\!\!\rightarrow} 363,130 \tag{35}$$
$$M_b = 677,880 \text{ in.-lb}$$

Using equation (6), we obtain

$$f_b = \frac{677,880}{76.02}$$

$$f_b = 8,920 \text{ psi}$$

Figure 53 shows the bearing reactions and critical section for the worst load condition (cable in, 30° forward) on the drum and support housing.

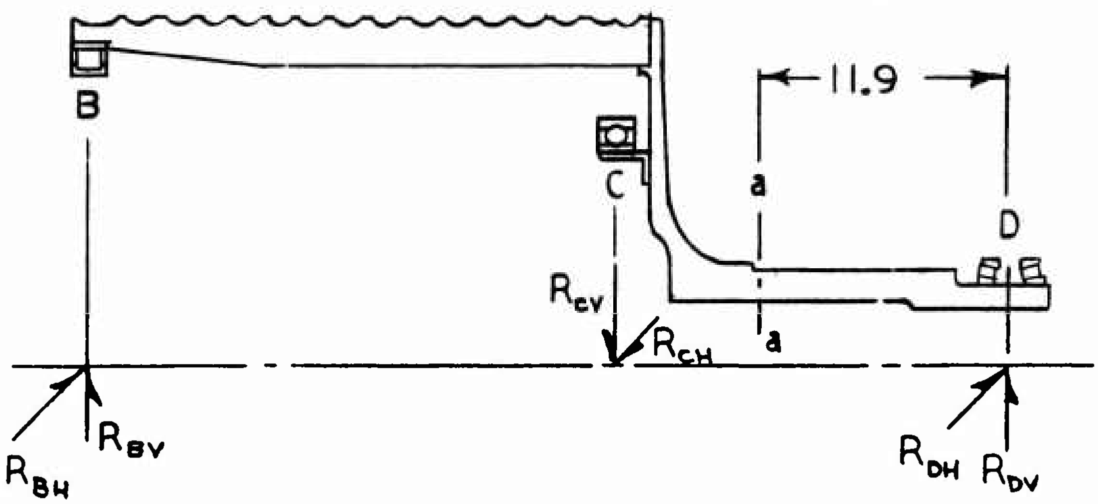


Figure 53. Drum and Support Housing, Single-Point Hoist.

At critical section a-a,

$$d_o = 14.2$$

$$d_i = 12$$

Using equation (32), we obtain

$$Z = \frac{\pi}{32} \frac{(14.2^4 - 12.0^4)}{14.2}$$

$$Z = 137.7$$

$$M_{bv} = 11.9 \; R_{Dv}$$

$$M_{bv} = 11.9 \;\; (93,510) \tag{36}$$

$$M_{bv} = 1,112,770$$

$$M_{bH} = 11.9 \;\; R_{DH}$$

$$M_{bH} = 11.9 \;\; (55,870)$$

$$M_{bH} = 664,850 \tag{37}$$

Using equation (35), we obtain

$$M_b = 1,112,770 \;+> 664,850$$

$$M_b = 1,296,250 \text{ in.-lb}$$

Using equation (6), we obtain

$$f_b = \frac{1,296,250}{137.7} = 9,410 \text{ psi}$$

$$f_b = 9,410 \text{ psi}$$

Mounting attachment loads

Figure 54 is a schematic drawing of the hoist housings showing the mounting feet locations.

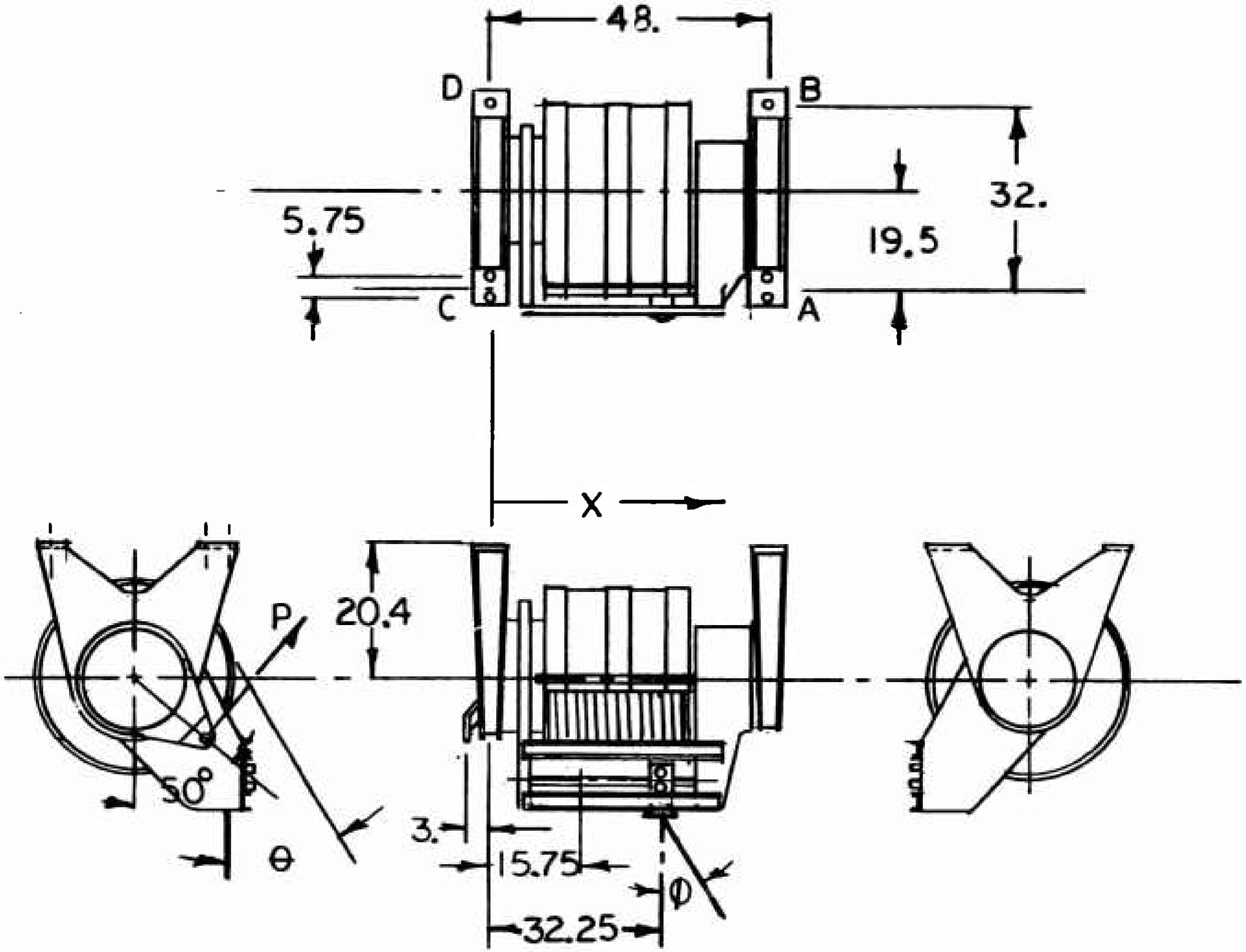


Figure 54. Mounting Arrangement, Single-Point Hoist.

From a statical equilibrium analysis of the system, the mounting foot loads are found to be

$$R_A = P_c\left[\frac{X \cos \phi}{122.9}(1.56 + \cos \theta - 1.632 \sin \theta) - .0821\right] \quad (38)$$

$$R_B = P_c\left[\frac{X \cos \phi}{78.8}(\cos \theta + 1.046 \sin \theta - 1) + .0345\right] \quad (39)$$

$$R_C = P_c\left[\frac{(48-X) \cos \phi}{122.9}(1.56 + \cos \theta - 1.632 \sin \theta) - 1.2360\right] \quad (40)$$

$$R_D = P_c\left[\frac{(48-X) \cos \phi}{78.8}(\cos \theta + 1.046 \sin \theta - 1) + .5178\right] \quad (41)$$

Foots A and C contain 2 bolts in each foot; hence, the bolt load is found by dividing the reactions at A and C in half. It is assumed that foots A and B share the shear loads. The foot shear loads are given by

$$S_A = S_B = \frac{P}{2}\sqrt{(\sin \theta \cos \phi + \sin 40^\circ)^2 + (\sin \theta \sin \phi)^2} \quad (42)$$

The worst case of bolt tension occurs in bolt D for cable out, zero side angle, 30° forward angle. Under these conditions the applied bolt load is 101,550 pounds. The total bolt load is given by

$$P = KP_a + P_i \quad (43)$$

where

P_a = applied load = 101,550 lb

P_i = initial bolt preload

K = pct of applied load felt in the bolt

For a flange thickness of 1.25 and a bolt diameter of .875,

K = .295

P_i = 30,000 lb(180,000 F_{tu} bolt, 7/8 diameter)

Using equation (43), we obtain

P = (.295) (101,550) + 30,000

P = 59,960 lb-Maximum bolt load

$P_{allowable}$ = 86,100 lb

Level Wind Mechanism

The level wind ball screw end loads are transmitted into the rigid drive half of the level wind arm. The two level wind arm bearings react all the moment caused by side cable loads and transfer the moment directly into the hoist side mounting plate. The worst case of stress in the level wind arm occurs under ultimate cable load conditions and 30° side angle. Figure 55 shows the level wind side arm and critical section.

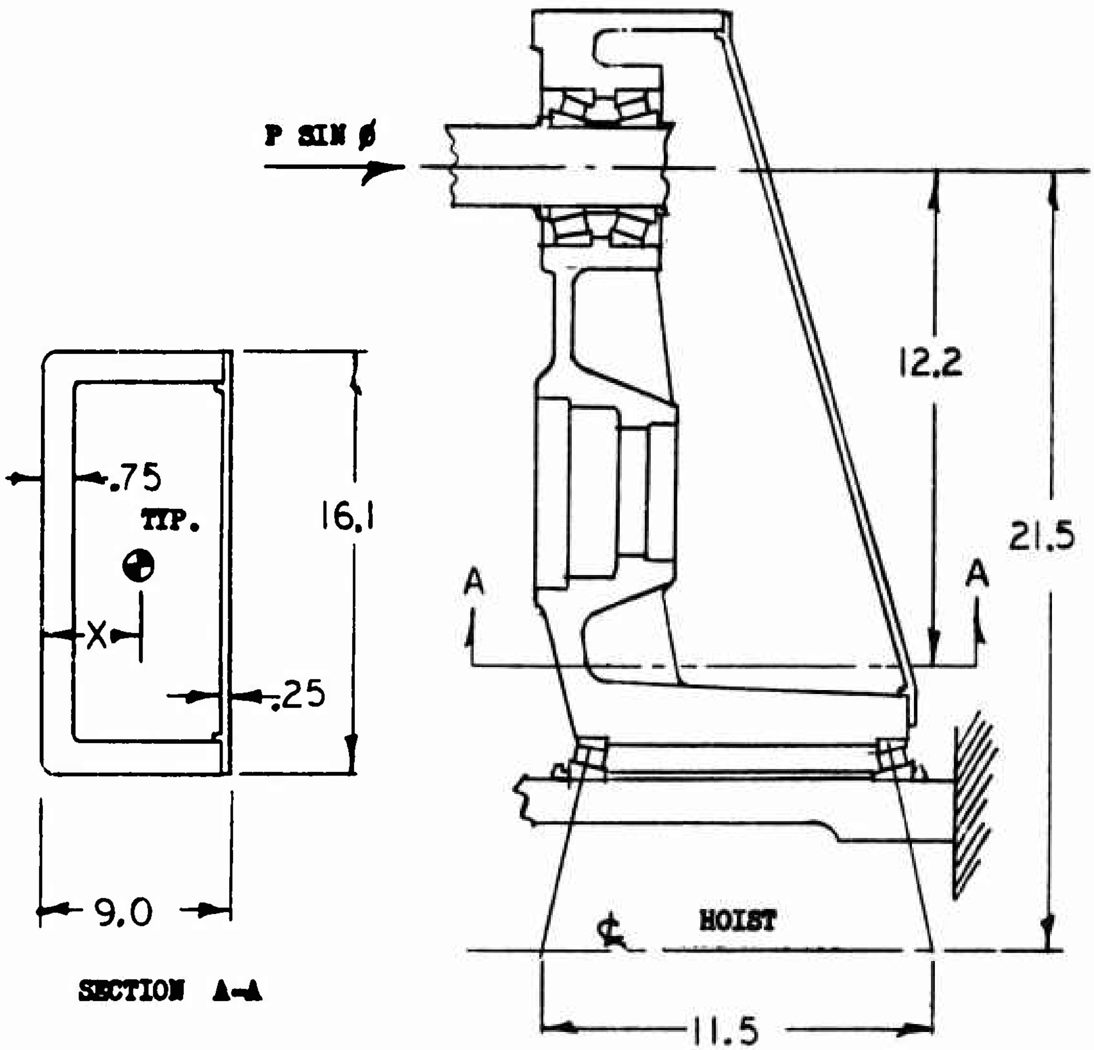


Figure 55. Critical Section, Level Wind Mechanism, Single-Point Hoist.

At critical section AA,

$$\bar{x} = 3.461$$

$$\overline{Ix} = 317$$

$$Z = 57.3$$

$$M_b = P (\sin \phi) 1$$

$$M_b = (150{,}000) (\sin 30^o) (12.2)$$

$$M_b = 915{,}000 \text{ in.-lb} \tag{44}$$

Using equation (6), we obtain

$$f_b = \frac{915{,}000}{57.3}$$

$$f_b = 15{,}970 \text{ psi}$$

For a 7079-T6 aluminum forging, $F_{tu} = 71{,}000$ psi

$$M.S. = \frac{F_{tu}}{f_b} -1$$

$$M.S. = \frac{71{,}000}{15{,}970} -1$$

$$M.S. = +3.45 \tag{45}$$

The bending moment on the hoist centerline is given by equation(44):

$$M_b = (150{,}000)(\sin 30^o) (21.5)$$

$$M_b = 1{,}612{,}500 \text{ in.-lb}$$

This moment is taken out by the two level wind ball bearings.

$$P_{brg} = \frac{M_b}{1}$$

$$P_{brg} = \frac{1{,}612.500}{11.5}$$

$$P_{brg} = 140,200 \text{ lb} \tag{46}$$

The static capacity (Co) of these bearings is 82,000 pounds; therefore,

$$P_{brg} = \frac{140,200}{82,000} = 1.71 \text{ Co} \tag{47}$$

$$\text{M.S.} = \frac{P_{allowable}}{P_{brg}} - 1 \tag{48}$$

Substituting in equation (48),

$$\text{M.S.} = \frac{3 \text{ Co}}{1.71 \text{ Co}} - 1$$

$$\text{M.S.} = +.75$$

The level wind bellmouth utilizes a screw to keep the cable tracking in the grooves of the drum. A ball screw is used to reduce friction and increase efficiency of the system. Reaction torque of the ball screw is provided by a fixed reaction pipe which also serves to stiffen the support structure. The basic ball screw data are given below:

Screw diameter	2-1/2
Ball diameter	3/8
Threads/inch	2
Lead	.5
No. of turns of balls	7
Operating load	35,280 lb (Reference 11)
Maximum static load	196,000 lb (Reference 11)

Since the drum pitch is 1.5 inches, the ratio from the drum to the screw is

$$RR = \frac{L_{screw}}{L_{drum}}$$

$$RR = \frac{.5}{1.5} = \frac{1}{3} \tag{49}$$

Hence the ball screw must turn three times faster than the drum. The axial loads felt by the screw are given by

$$P_a \text{ (normal)} = P_{(normal)} \sin \phi \text{ max}$$

$$P_a \text{ (normal)} = 4000 (\sin 30^\circ) = 20{,}000 \text{ lb} \tag{50}$$

$$P_a \text{ (max)} = P_{max} \sin \phi \text{ max}$$

$$P_a \text{ (max)} = 150{,}000 (\sin 30^\circ) = 75{,}000 \text{ lb} \tag{51}$$

These loads are below the allowable loads for the ball screw as given in Reference 11 for 1 million inches of ball travel. For 12 wraps of cable, 1 million inches of travel is equivalent to 27,800 cycles of hoist operation.

The screw must also be designed to avoid buckling as a column.

$L = 29.5$ (distance between supports)

d_o (effective) $= 2.31$

$d_i = 1.75$

$I = .937 \text{ in.}^4$

The critical buckling load for pinned ends is given by

$$P_{cr} = \frac{\pi^2 EI}{L^2}$$

$$P_{cr} = \frac{\pi^2 (30 \times 10^6)(.937)}{(29.5)^2}$$

$$P_{cr} = 318{,}800 \text{ lb} \tag{52}$$

The torque required to turn the screw against the axial load P_a is given by

$$T = \frac{P_a l}{2 \pi \eta} \tag{53}$$

The efficiency η is conservatively assumed to be 90 pct (Reference 8):

$$T_{normal} = \frac{20,000\ (.5)}{2\ \pi(.9)}$$

$$T_{normal} = 1,770 \text{ in.-lb (at level wind ball screw)}$$

The scrub roll is a rubber coated roller that pulls the cable off the drum when lowering and is free wheeling when raising the cable. The cable is loaded against the scrub drive roller by the scrub roll pulley which is adjustable to compensate for wear and to obtain the proper initial tension. The scrub roll pulley is mounted on the bellmouth and travels along the ball screw, keeping the same relative position with the cable as shown in Figure 56.

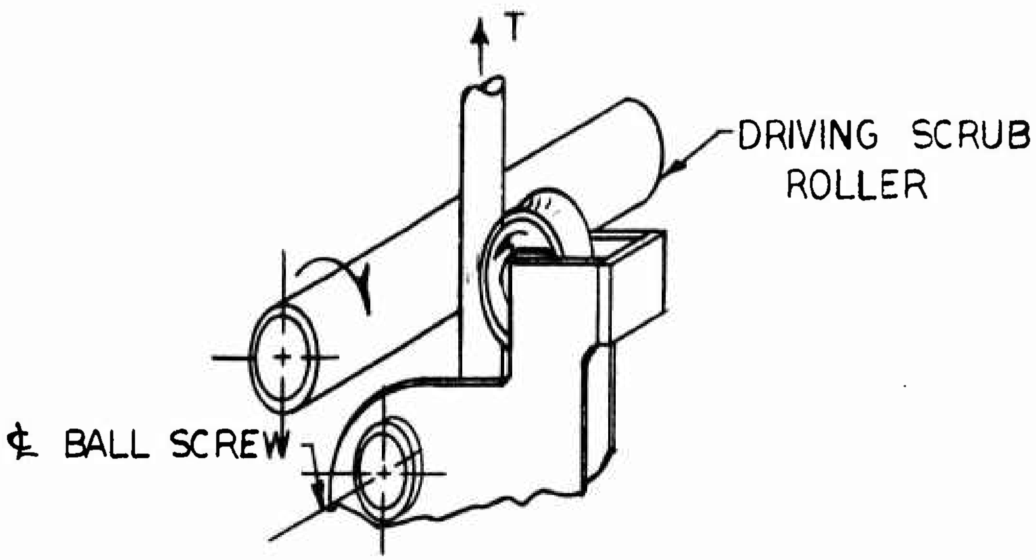


Figure 56. Bellmouth, Scrub Roller, and Ball Screw Assembly, Single-Point Hoist.

To provide a minimum tension of 50 pounds in the cable at all times, the normal force required on the pulley is found by

$$F_n = \frac{T}{\mu} \qquad (54)$$

Assuming that the coefficient of friction is .3, we obtain

$$F_n = \frac{50}{.3}$$

$$F_n = 167 \text{ lb}$$

This force produces a torque on the scrub roller given by

$$T = F_n \frac{d}{2}$$

$$T = 167 \frac{2.75}{2}$$

$$T = 230 \text{ in.-lb (torque required to drive scrub rolls)} \qquad (55)$$

Weston Brake

The Weston brake is used to raise the load with all units locked, to hold the load with the input power removed, and to lower the load at the same speed as the motor. The important design considerations are adequate heat dissipation in the discs when lowering the load, the proper lead angle of the unlocking device to prevent self-locking if the angle is too low, and failure to lock if the angle is too high. For heat dissipation, the plate pressure must be low.

d_o = 9.00

d_i = 6.00

n = 7 friction surfaces

T = 6430 in.-lb (40,000-lb cable load)

The brake discs are SAE 1095 steel against high friction bronze. For these materials operating in oil, the coefficient of friction is 0.07.

$$P_a = \frac{8\,T}{\pi \mu d_i (d_o^2 - d_i^2)\, n}$$

$$P_a = \frac{8(6430)}{\pi (.07)(6.00)(9.00^2 - 6.00^2)(7)}$$

$$P_a = 96.3 \text{ psi} \qquad (56)$$

P_a (allowable) = 150 psi

Screw Data

$5 - 1\frac{1}{3}$ triple thread

$$\text{lead} = l = 2.25$$

$$d_m = 4.25$$

The lead angle is given by

$$\alpha = \text{arc tan } \left(\frac{l}{\pi d_m}\right)$$

$$\alpha = \text{arc tan } \frac{2.25}{\pi (4.25)}$$

$$\alpha = \text{arc tan } .16851$$

$$\alpha = 9^\circ 34' \qquad (57)$$

For proper operation (Reference 7),

$$6^\circ \leq \alpha \leq 12^\circ \qquad (58)$$

Heat Rise in Weston Brake

When the Weston brake is used to lower the load, all the energy of the load must be absorbed by the brake. This energy is the work done per revolution times the number of revolutions required to lower the load. Assuming that a 40,000 pound is being lowered 100 feet, we obtain

$$t = \frac{L}{V} \qquad (59)$$

where

t = time to lower load, min

L = length of cable, ft

V = cable velocity, fpm

$$t = \frac{100}{60.2} = 1.661 \text{ min}$$

Heat generated is given by

$$H = \frac{E_k}{778} \text{ Btu} \tag{60}$$

where

E_k = energy expended in ft-lb

$$E_k = (2\pi T)(\text{rpm})(t) \tag{61}$$

where

T = torque = 536 ft-lb

rpm = 716

$$E_k = 2\pi(536)(716)(1.661)$$

$$E_k = 4{,}010{,}000 \text{ ft-lb}$$

$$H = \frac{4{,}010{,}000}{778}$$

H = 5,150 Btu per Weston brake lowering operation

The temperature rise after one lowering operation is

$$\Delta t = \frac{H}{C_b W_b + Co\, W_o} \tag{62}$$

where

Δt = temperature rise, °F

C_b = specific heat of brake parts = .12 Btu/lb/°F

W_b = weight of brake parts that will heat up, lb

Co = specific heat of oil = .55 Btu/lb/°F

W_o = weight of oil surrounding brake, lb

While the drive train and surrounding castings weigh 589 pounds, it has been conservatively assumed that 150 pounds of metal and 25 pounds of oil which surround the Weston brake are the effective heat sink during a lowering operation. Using equation (62), we obtain

$$\Delta t = \frac{5150}{(.12)(150) + (.55)(25)}$$

$$\Delta t = 162\ ^{o}F$$

The temperature rise is 162°F in one braking operation. Between operations, the plates will cool by convection and radiation.

Free Reeling Clutch

The free reeling clutch is normally engaged by the axial load caused by the compression of a stack of Belleville washers. Since this clutch never slips, it is designed on static torque capacity and can have a high plate pressure. In addition, there will be very little oil on the plates and the friction torque will be governed by the static coefficient of friction. The larger Timken bearing carries the axial clutch load during normal operation but has no relative rotation between inner and outer races. The ball bearings used to prevent rotation of the hydraulic cylinder rotate during normal operation but carry no load. When the free reel clutch is disengaged, these bearings must withstand the axial load created by the hydraulic cylinder necessary to compress the Belleville washers and free the clutch plates.

During free fall of the load, high rotational speeds are produced, causing dynamic tensile stresses in rotating parts. An analysis is made showing the effect of inertia on final speed after free fall and also the effect of stresses in critical parts due to the high rotational speed.

$$T_{design} = T_{normal} \times 3.75 \times F.S. \qquad (63)$$

Assuming a factor of safety of 1.25, we obtain

$$T_{design} = 6430 \times 3.75 \times 1.25$$

$$T_{design} = 30{,}140 \text{ in.-lb}$$

$$d_o = 6.00$$

$$d_i = 4.00$$

$$n = 11 \text{ friction surfaces}$$

$$\mu = .25 \text{ (static)}$$

Using equation (56), we obtain

$$P_a = \frac{8\ (30{,}140)}{\pi\ (.25)(4.00)6.00^2 - 4.00^2)\ 11}$$

$P_a = 348$ psi

The axial clutch load necessary to produce this pressure is

$$F_a = \frac{\pi \, P_a \; d_i(d_o - d_i)}{2}$$

$$F_a = \frac{\pi \,(348)(4.00)(6.00 - 4.00)}{2}$$

$$F_a = 4370 \text{ lb} \qquad (64)$$

Belleville Washer Design (Reference 15)

The Belleville washer must provide a preload of 4370 pounds on the clutch plates for proper torque. This load must be produced before the spring becomes flat so that further compression can take place in order to release the clutch plates. The stack of Belleville washers is shown in Figure 57.

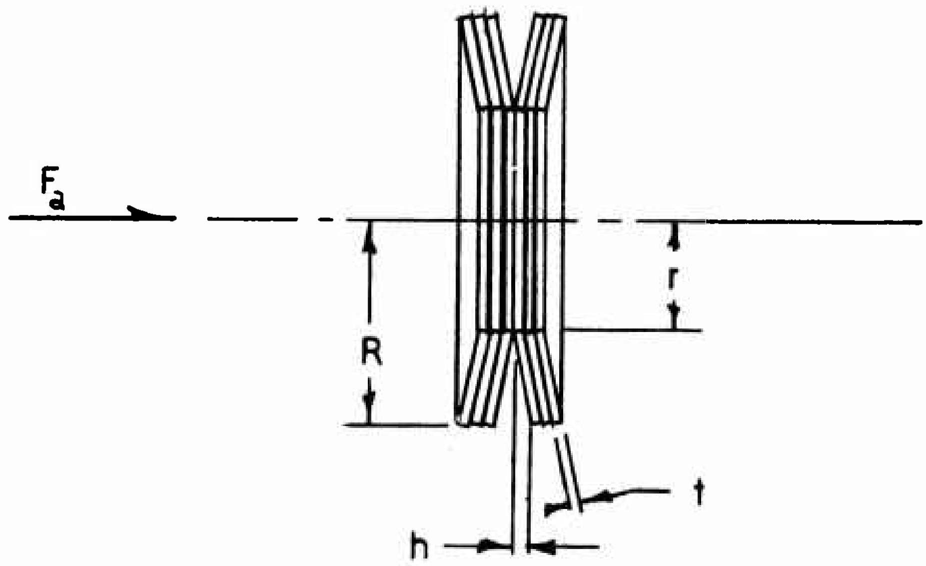


Figure 57. Belleville Washers, Free Reeling Clutch, Single-Point Hoist.

$$R = 3.00$$

$$r = 2.00$$

$$t = .120$$

$$h = .156$$

$$F_a = \frac{C_1\ C\ E\ t^4}{R^2} \qquad (65)$$

where

C_1 is a function of δ/t and h/t and is given by a curve in Reference 15.

C is a function of R/r and is given by a curve in Reference 15.

With the values of $R/r = \frac{3.00}{2.00} = 1.5$ and $C = 1.94$ substituted in equation (65), we obtain

$$C_1 = \frac{4370\ (3.00)^2}{1.94(30 \times 10^6)(.12)^4}$$

$$C_1 = 3.26$$

For three springs in parallel,

$$C_1 = \frac{3.26}{3} = 1.09$$

For this C_1 and $\frac{h}{t} = 1.3$, $\frac{\delta}{t} = .7$,

$$\delta = .7t = .084$$

For two springs in series,

$$\Delta = 2\,\delta \qquad (66)$$

$$\Delta = (2)(.7)(.120) = .168$$

The amount left for compression to release the clutch is

clutch plate clearance $= 2h - \Delta$

$$\text{clutch plate clearance} = 2(.156) - .168$$

$$\text{clutch plate clearance} = .144 \qquad (67)$$

To find the force required to flatten the springs and release the clutch,

$$\frac{\delta}{t} = \frac{h}{t} = 1.30 \qquad (68)$$

$$C_1 = 1.42$$

For three springs in parallel,

$$C_1 = 3(1.42)$$

$$C_1 = 4.26$$

With these conditions the force required to flatten the springs is given by equation (65):

$$F_a = \frac{4.26(1.94)(30 \times 10^6)(.12)^4}{(3.00)^2}$$

$$F_a = 5710 \text{ lb} - \text{Force required to flatten springs}$$

Drum Speed After Free Fall of Load

The isolated drum and load are shown in Figure 58. The mass polar moment of inertia, J, of the drum includes all rotating internal parts from the drum to the free reeling clutch. These parts are related to the speed of the drum by the ratio of the speed squared.

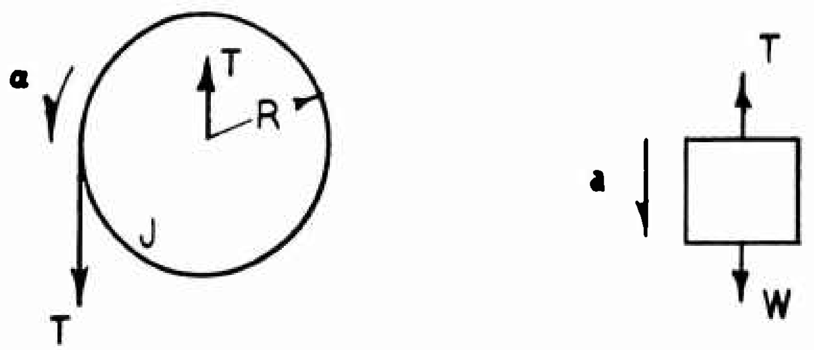


Figure 58. Free Falling of Load, Single-Point Hoist.

$$\Sigma F = ma \tag{69}$$

$$W - T = \frac{W}{g} a \tag{70}$$

$$\Sigma M = J \alpha \tag{71}$$

$$TR = J\alpha = J \frac{a}{R} \tag{72}$$

Here, the acceleration is due to gravity: $a = g$

Also,

$$a = R \omega T \tag{73}$$

$$S = \frac{1}{2} a t^2 \tag{74}$$

Solving for ω and eliminating time, t, and cable tension, T,

$$\omega = \sqrt{\frac{2 W S}{R^2 \quad \frac{W}{g} + \frac{J}{R^2}}} \tag{75}$$

$$rpm_{drum} = \frac{30}{\pi} \sqrt{\frac{2 W S g}{W R^2 + g J}} \tag{76}$$

final drum speed after fall of weight W through distance S.

$$J = 4736 \text{ in.-lb/sec}^2$$

for all rotating parts of the single-point hoist from the drum to the drum to the free reel clutch and reflected to the drum speed.

for $R = 17$, $g = 386 \text{ in./sec}^2$, and $J = 4736 \text{ in.-lb/sec}^2$,

$$rpm_{drum} = \frac{30}{\pi} \sqrt{\frac{2 W S (386)}{W (17)^2 + 386 (4736)}}$$

$$rpm_{drum} = 9.549 \sqrt{\frac{2.67 \ W \ S}{W + 6326}}$$

Figure 59 shows a plot of drum speed increase factor versus distance of fall for various weights.

If $J = 0$, the equation for RPM of the drum reduces to

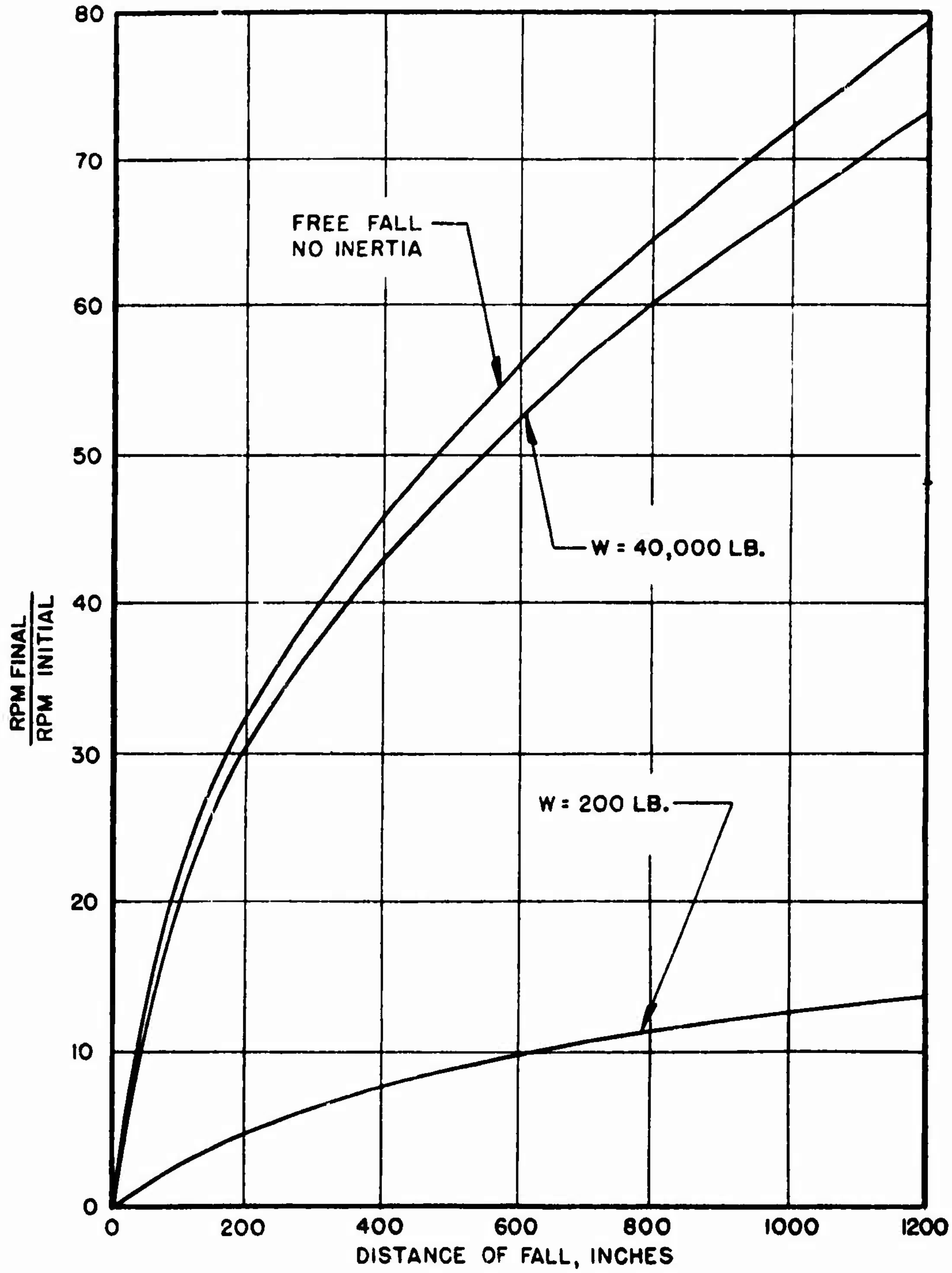


Figure 59. Load vs Free Fall Velocity, Single-Point Hoist.

$$rpm_{drum} = \frac{30}{\pi} \sqrt{\frac{2\,S\,g}{R^2}}$$

$$rpm_{drum} = 9.549 \sqrt{2.671\ S} \qquad (77)$$

As can be seen in Figure 59, when heavy loads are on the hoist and the free wheel unit is released, the drum inertia has little effect on the final rotational speed over that of a free falling body.

The stress due to high rotational speed on a cylinder of constant thickness (Reference 6) is

$$f_{r\,max} = \frac{(3+\nu)}{32} \frac{\rho\omega^2}{g} (d_o - d_i)^2 \qquad (78)$$

$$f_{t\,max} = \frac{\rho\,\omega^2}{16\,g} \left[(3+\nu)\, d_o^2 + (1-\nu)\, d_i^2 \right] \qquad (79)$$

The free reeling clutch shaft is normally turning at 716 rpm. After free fall of 100 feet with a 40,000-pound cable load, the shaft is turning at 53,104 rpm. The free reeling clutch output side reaction plate is the most highly stressed member under these conditions.

For this plate,

$$d_o = 6.00$$

$$d_i = 0$$

$$\rho = .283 \text{ lb/in.}^3$$

$$\nu = .3$$

$$\omega = rpm \times \frac{\pi}{30}$$

$$\omega = 53{,}104 \frac{\pi}{30}$$

$$\omega = 5{,}561 \text{ rad/sec} \qquad (80)$$

Using equation (78), we obtain

$$f_{r\,max} = \frac{(3+.3)}{32} \frac{(.283)(5561)^2}{386} (6.00 - 0)^2$$

$$f_{r\ max} = 84{,}160 \text{ psi}$$

Using equation (79), we obtain

$$f_{t\ max} = \frac{(.283)(5561)^2}{16\ (386)} \left[(3 + .3)(6.00)^2 + (1 - .3)\ (0)^2 \right]$$

$$f_{t\ max} = 168{,}300 \text{ psi}$$

The free reeling clutch output plate is made from AMS 5000 steel whose ultimate tensile strength is 200,000 psi and whose yield strength is 176,000 psi. The maximum tensile stress produced in this part by high speed rotation is therefore below the yield strength.

Gear Design

The face width of the spur gears in the single-point hoist may be governed by bending stress or Hertz stress. The formulas for these stresses are as follows:

$$f_b = \frac{1.5}{X} \frac{W_t}{F} \frac{K_t}{} \tag{81}$$

$$f_c = \sqrt{\frac{21 \times 10^6\ W_t}{\sin 2\ \phi\ F} \left(\frac{1}{d_p} \pm \frac{1}{d_g} \right)} \quad \begin{matrix} (+ \text{ for external gear}) \\ (- \text{ for internal gear}) \end{matrix} \tag{82}$$

The tangential tooth load, W_t, is given by

$$W_t = \frac{2\ T}{d} \tag{83}$$

The torque, T, is found by dividing the torque at the drum by the appropriate gear ratio (from the drum to the gear in question):

$$T = \frac{P_c\ d_m}{2\ RR} \tag{84}$$

Table XXIV summarizes the bending and compressive stresses for all the spur gear teeth on the single-point hoist. The tangential tooth loads on the level wind gears are derived by finding the torque required to turn the level wind feed screw to overcome the axial load caused by a 30° side cable angle. Since the gear tooth ultimate tensile strength is greater than 3.75 x bending stress allowable, the lowest margins of safety occur during normal operating conditions; hence the ultimate bending stresses have been omitted.

Shaft bearing reactions caused by gear loads may be found by well-known

TABLE XXIV
GEAR SUMMARY - SINGLE-POINT HOIST

	N No. of Teeth	Pitch	d Dia.	F Face Width	X Tooth Form Factor	Kt Stress Conc. Factor	Wt Tang. Tooth Load @ P=40,000 lb	fb Bending Stress	fc Hertz Stress
Input Spur Pinion	21	10	2.100	2.00	.0857	1.16	1460	14,820	138,950
Input Spur Idler	33	10	3.300	1.75	.0986	1.18	1460	14,980	138,950
Input Spur Gear	88	10	8.800	.90	.1250	1.25	1460	24,330	141,680
2nd Stage Spur Pinion	39	10	3.900	2.00	.1040	1.19	3300	28,320	131,040
2nd Stage Spur Gear	137	10	13.700	1.88	.1282	1.28	3300	26,290	131,040
Planetary Sun Gear	72	7	10.286	.50	.1722	1.07	550	10,240	94,300
Planetary Main Pinion	40	7	5.714	2.55	.1493	1.01	7830	31,160	130,900
Planetary Output Ring Gear	152	7	21.714	1.75	.247	1.15	7830	31,250	130,900
Planetary Secondary Pinion	36	7.25	4.966	2.55	.1396	1.02	7280	31,290	137,780
Planetary Fixed Ring Gear	152	7.25	20.966	1.75	.238	1.16	7280	30,410	137,780
Level Wind Drive Pinion	165	10	16.500	.44	.1295	1.29	644	21,870	81,150
Level Wind Idler	105	10	10.500	.50	.1275	1.27	644	19,240	102,140
Ball Screw Drive Gear	55	10	5.500	.65	.1142	1.21	644	15,750	102,140

methods, as shown in Reference 3. All bearing loads, ratings, and lives are tabulated in Tables XXV and XXVI for normal and static conditions.

Drive System Shafting

The drive train shafts may be subjected to bending stresses, torsional stresses, or a combination of both. In bending, the hoist shafts are critical under normal load fatigue conditions, since the endurance limit multiplied by the ultimate load factor is less than the ultimate tensile strength of the material. In torsion, however, the shafts are critical under ultimate load conditions. The single-point hoist shafting has been analyzed by the methods of Reference 3 and the results are given in Table XXVII.

Cable

The cable for the single-point hoist is designed for a 150,000-pound ultimate load. This load is derived by the following method.

$$P_{ult} = P_{limit} \times F.S. \tag{85}$$

$$P_{limit} = P_{normal} \times (G\ Factor) \tag{86}$$

For the single-point hoist, the G Factor is 2.5 and the factor of safety for ultimate load conditions is 1.5.

$$P_{limit} = 40{,}000\ (2.5) = 100{,}000\ lb$$

$$P_{ult} = 100{,}000\ (1.5) = 150{,}000\ lb$$

The cable breaking load is given by

$$P = F_{tu}\ K\ A \tag{87}$$

where

K = Area Factor = .71

A = Total Area of all Strands

For the single-point hoist, the wire diameter is .060 in. and the number of wires is 342 (18 x 19 construction)

$$A = \text{No. of wires}\ \frac{\pi}{4}\ d^2$$

TABLE XXV
SUMMARY OF BEARING LIVES AND LOADS - SINGLE-POINT HOIST

Bearing Location	RPM	Bearing Load at 40,000 lb Cable Load (lb)		Bearing Load at 100,000 lb Cable Load (lb)		Bearing Static Allowable Load (lb)	Bearing Dynamic Capacity (lb)	Life (hr)
		Thrust	Radial	Thrust	Radial			
Input Pinion, Input End	3000.0	0	790	-	-	-	5,360	1,735
Input Pinion, Output End	3000.0	0	790	-	-	-	5,360	1,735
Input Idler, Input End	1909.0	0	1,460	-	-	-	6,690	840
Input Idler, Output End	1909.0	0	1,460	-	-	-	6,690	840
Input Gear, Input End	716.0	0	860	-	-	-	5,670	6,670
Input Gear, Output End	716.0	0	720	-	-	-	7,330	24,560
2nd Stage Pinion, Input End	716.0	0	1,790	0	4480	11,750	12,600	8,120
2nd Stage Pinion, Output End	716.0	0	1,790	0	4480	11,750	12,600	8,120
2nd Stage Gear, Input End	203.8	0	3,080	0	7700	12,500	12,600	5,600
2nd Stage Gear, Output End	203.8	0	500	0	1250	12,500	12,600	1,308,000
Compound Plan. Pinion (right)	257.2	0	5,340	0	13350	15,900	15,700	1,650
Compound Plan. Pinion (left)	257.2	0	5,480	0	13700	15,700	15,700	1,520

Notes: 1. Static loads are not felt from the input to the Weston brake.
2. Any bearings not shown carry no load (or negligible load).

TABLE XXVI

SUMMARY OF BEARING LIVES AND LOADS - SINGLE-POINT HOIST

Bearing Location	Ref. Note No.	RPM	Bearing Load @ 40,000 lb Cable Load (lb) Thrust	Bearing Load @ 40,000 lb Cable Load (lb) Radial	Bearing Load @ 100,000 lb Cable Load (lb) Thrust	Bearing Load @ 100,000 lb Cable Load (lb) Radial	Bearing Static Allow. Load (lb)	Bearing Dynamic Capacity (lb)	Life (hr)
Free Reel Clutch									
Main Thrust	-	0.0	4,370	0	4,370	0	*52,100	-	
Piston Isolator (pair)	1,3	716.0	5,710	0	5,710	0	*36,100	-	
Washer Preload	1,3	716.0	5,710	0	5,710	0	*18,700	-	
Level Wind									
Idler, Gear End	2,4	10.1	0	1,740	0	4,350	6,850	6,580	88,800
Idler, Outboard End	4	10.1	0	450	0	1,130	3,680	3,770	965,500
Ball Screw, Double Timken	4	20.3	20,000	700	50,000	1,750	*50,800	17,000	410
Ball Screw, Reaction End	4	20.3	0	320	0	800	4,780	4,890	2,930,000
Support	4	0.0	-	-	50,000	117,000	246,000	-	
Main Support (Brg. A, Fig. 50)	5	0.0	-	-	0	133,500	516,000	-	
Main Drum (Brg. B, Fig. 50)	6	6.8	0	42,680	0	106,700	230,000	100,000	41,860
Secondary Drum (Brg. C, Fig. 50)	6	6.8	0	18,260	0	45,650	100,000	49,500	49,060
Main Thrust (pair) (Brg. D, Fig. 50)	7	6.8	20,000	14,530	50,000	36,340	67,250	16,980	270

Notes:
1. Free reel bearings are loaded only when the free reel piston is activated.
2. Angular load positions exist only momentarily. The lives calculated in the above table assume that angular conditions exist constantly.
3. Free reel piston on.
4. 30^{o} side load of cable.
5. Cable out, 60^{o} aft.
6. Cable out, 30^{o} forward.
7. Cable in. 0^{o} forward, 30^{o} to the side.
8. Any bearings not shown carry no load (or negligible load).

*Thrust load

TABLE XXVII
CRITICAL SECTION SHAFT STRESSES - SINGLE-POINT HOIST

Location	Critical Section	Type of Critical Stress	Stress (psi)	Margin of Safety
Input Pinion Shaft	Input Spline Undercut	Torsion	13,300	5.54
Input Idler Shaft	Bearing Radius	Bending	7,530	1.66
Input Gear Shaft	Spline Undercut	Bending	1,810	10.0
Free Reel Clutch Input	Torque Shaft	Torsion	30,900	1.82
Free Reel Clutch Output	Torque Shaft	Torsion	60,700	.43
2nd Stage Pinion Shaft	Bearing Radius	Torsion	11,000	6.90
2nd Stage Gear Shaft	Bearing Radius	Torsion	3,600	23.2
Compound Planetary Pinion	Between Gears	Torsion	3,250	25.8
Output Ring Gear	Torque Shaft	Torsion	7,180	11.1
Level Wind Idler	Gear End Bearing Radius	Bending	1,800	10.1

Note: Torsional stresses are critical under ultimate load conditions.

$$A = 18 \times 19 \times \frac{\pi}{4} (.060)^2$$

$$A = .967 \text{ in.}^2 \qquad (88)$$

$$F_t = 250{,}000 \text{ psi}$$

$$P = (250{,}000)(.71)(.967)$$

$$P = 171{,}600 \text{ pounds - cable breaking strength}$$

SINGLE-POINT HOIST DRIVE SYSTEM

Clutch-Reverser Unit

The clutch-reverser unit is the power source and directional control for the single-point hoist. Power is taken from the accessory gearbox to which the clutch-reverser unit is mounted. The 1.364 to 1 reduction gearbox consists of three mutually geared shafts, two of which contain hydraulic clutches as shown in Figure 72. One clutch is used for raising the load while the other is used for lowering the load. The hydraulic control system prevents both clutches from simultaneously engaging.

Design Data

Input rpm	=	5995.2
Output rpm	=	4396.5
Reduction ratio	=	1.364 to 1
Input torque	=	957 in.-lb (40,000-lb cable load)
Horsepower	=	91

The gear tooth bending and compressive stresses are found by methods similar to those used in the single-point hoist and are summarized in Table XXVIII, page 193.

Bearing loads and lives are found by methods similar to those of Reference 3. All bearing loads, ratings, and lives are tabulated in Table XXIX, page 194, for the clutch-reverser unit.

Clutch Analysis

Both the load lifting and load lowering clutches of the clutch-reverser unit are identical in design. The load lowering clutch can have a much smaller capacity than the load lifting clutch, since it must only overcome friction in the system. The units are made identical because of cost and assembly reasons, since their weight is small.

The clutches are hydraulically actuated by a supply pressure of 250 psi. The axial load is given by

$$F_a = P\,A \tag{89}$$

where

P = pressure = 250 psi

A = piston area = 7.903 in.^{2}

$$F_a = 250\ (7.903)$$

$$F_a = 1980 \text{ lb}$$

The torque capacity is given by

$$T = \frac{F_a}{4}\ (d_o + d_i)n \tag{90}$$

where

d_o = plate outside diameter = 3.82

d_i = plate inside diameter = 2.00

n = number of friction surfaces = 16

μ = coefficient of friction = .07 dynamic (Reference 5)

$$T = \frac{1980\ (.07)}{4}\ (3.82 + 2.00)(16)$$

$$T = 3220 \text{ in.-lb}$$

On the low-speed (4396.5 rpm) shaft, this torque is equivalent to 224 horsepower which is well above the 94 horsepower design criteria even when using the very conservative coefficient of friction of .07.

Using equation (56), the plate pressure at a torque of 3220 in.-lb is

$$P_a = \frac{8\ (3220)}{\pi\ (.07)(2.00)(3.82^2 - 2.00^2)\ 16}$$

$$P_a = 366 \text{ psi}$$

The clutch plates are lubricated by a constant oil supply and only slip during acceleration of the load.

TABLE XXVIII

SUMMARY OF GEAR TOOTH BENDING AND COMPRESSIVE STRESSES-CLUTCH-REVERSER UNIT

	N No. of Teeth	Pitch	d Diameter	F Face Width	X Tooth Form Factor	K_t Stress Conc. Factor	Weight Tang. Tooth Load at P= 40,000 lb	f_b Bending Stress	f_c Comp. Stress
Input Drive Pinion	44	10	4.400	.27	.1075	1.19	435	26,750	137,300
Reversing Shaft Gear	60	10	6.000	.27	.1160	1.21	435	25,210	137,300
Reversing Shaft Pinion	60	12	5.000	.44	.0966	1.28	521	23,530	116,800
Main Drive Pinion	44	12	3.667	.44	.0896	1.26	521	24,980	128,900
Output Gear	60	12	5.000	.50	.0966	1.28	521	20,710	128,900

TABLE XXIX
SUMMARY OF BEARING LOADS AND LIVES-
CLUTCH-REVERSER UNIT

Location	Worst Case Bearing Radial Load at 94 HP	C Dynamic Capacity	RPM	Life (Hours)
Input Shaft, Gear End	424	2760	5995.2	767
Input Shaft, Outboard End	183	1100	5995.2	604
*Input Shaft, Clutch Drive Gear End	282	2900	0	-
*Input Shaft, Clutch Drive Clutch End	282	2900	0	0
Reverser Shaft, Gear End	424	2290	4396.5	597
Reverser Shaft, Outboard End	183	1100	4396.5	823
*Reverser Shaft, Clutch Drive Gear End	282	2900	0	-
*Reverser Shaft, Clutch Drive Clutch End	282	2900	0	-
Output Shaft, Gear End	443	2760	4396.5	917
Output Shaft, Outboard End	121	2760	4396.5	44,990

*Note: These bearings have no relative rotation when loaded. When the opposite clutch is driving, the relative rotation is twice normal but the bearing load is zero.

Upper Angle Gearbox

The upper angle gearbox (Figure 72) is mounted directly on the output of the clutch-reverser unit. The 1.172 to 1 reduction ratio is accomplished by a single set of spiral bevel gears which turn the drive shaft downward and sideward toward the hoist angle gearbox.

Spiral Bevel Gear Data

α	=	Shaft angle	=	$139^\circ 30'$
ϕ	=	Pressure angle	=	20°
ψ	=	Spiral angle	=	35°
P_d	=	Diametral pitch	=	6
N_p	=	Number of teeth in pinion	=	29
d_p	=	Pitch diameter of pinion	=	4.833
N_g	=	Number of teeth in gear	=	34
d_g	=	Pitch diameter of gear	=	5.667
F	=	Face width	=	.86
M_p	=	Contact ratio	=	1.35

Table XXX summarizes bearing lives for the four tapered roller bearings in the reverser output angle gearbox for 40,000-pound cable load.

TABLE XXX
BEARING LIVES AND LOADS-
UPPER ANGLE GEARBOX

Location	Loads (Pounds) Thrust	Radial	Radial Equivalent	Basic Radial Rating	Life (Hours)
Input Pinion, Gear End	483	1070	1455	1800	662
Input Pinion, Input End	0	238	711	865	700
Output Gear, Gear End	216	1110	1356	1240	1020
Output Gear, Output End	0	400	400	510	5250

Lower Angle Gearbox

The lower angle gearbox (Figure 72) is mounted on the airframe near the hoist input. A drive shaft connects the input to the output of the reverser angle gearbox, while another drive shaft connects the output to the hoist input pinion. The 1.25 to 1 reduction ratio is accomplished by a single set of spiral bevel gears.

Spiral Bevel Gear Data

α	=	Shaft angle	=	80°
ϕ	=	Pressure angle	=	20°
ψ	=	Spiral angle	=	35°
P_d	=	Diametral pitch	=	6
N_p	=	Number of teeth in pinion	=	20
d_p	=	Pitch diameter of pinion	=	3.667
N_g	=	Number of teeth in gear	=	25
d_g	=	Pitch diameter of gear	=	4.167
F	=	Face width	=	.88
M_p	=	Contact ratio	=	1.40

Table XXXI summarizes bearing lives for the four tapered roller bearings in the hoist input angle gearbox for 40,000-pound cable load.

TABLE XXXI
BEARING LIVES AND LOADS-
LOWER ANGLE GEARBOX

Location	Loads (Pounds) Thrust	Radial	Radial Equivalent	Basic Radial Rating	Life (Hours)
Input Pinion, Gear End	895	1625	2852	12,020	540
Input Pinion, Input End	0	705	1646	7,870	800
Output Gear, Gear End	875	1810	3140	11,910	476
Output Gear, Output End	0	873	1329	6,140	903

FOUR-POINT HOIST

The main drive train of the four-point hoist consists of four stages of gearing as shown in Figure 73, page 261. A hydraulic motor drives the first stage spur pinion, idler, and gear of reduction ratio 4.517 to 1. The first stage gear shaft drives the input section of the Weston brake. As in the single-point hoist, the Weston brake turns as a unit when raising the load, slips at the same speed as the input when lowering the load, and holds the load with the input power off. A hydraulically powered free reel clutch similar to that used in the single-point hoist permits the hook and cable assembly to be jettisoned in the event of a hook release malfunction. It is located between the Weston brake and the second stage spur pinion. The second stage of gearing is a 6.273 to 1 reduction ratio spur pinion gear. A conventional planetary (sun gear driving, ring gear fixed, cage output) third stage of 4.903 to 1 reduction ratio is driven from the second stage gear. The output cage of the third stage planetary drives the sun gear of the 3.702 to 1 reduction ratio fourth stage conventional planetary whose output cage is bolted directly to the drum.

The level wind ball screw is rotated by a chain drive whose drive sprocket is attached to the drum side plate. The main support is a two-way swivel arrangement which does not allow the transmission of any moment loading into the airframe. A hydraulic load isolator is mounted between the airframe and hoist to dampen load oscillations.

Design Data

Normal cable load	=	11,550 lb
Limit cable load	=	32,300 lb
Yield cable load	=	37,200 lb
Ultimate cable load	=	48,500 lb
Cable diameter	=	.79 in.
Useful cable length	=	50 ft
Mean drum diameter	=	22.19 in.
Cable angle (static)	=	$\pm 30°$ (any direction)
Cable angle (dynamic)	=	$\pm 15°$ (any direction)
Input speed	=	2750 rpm
Overall reduction ratio	=	514.356
1st Stage spur ratio	=	4.517
2nd Stage spur ratio	=	6.273
3rd Stage planetary ratio	=	4.903
4th Stage planetary ratio	=	3.702
Cable pitch line velocity	=	31.06 fpm

Drum

The drum for the four-point hoist has a mean diameter of 22.19 inches and a length of 10.5 inches. The analysis of this component is summarized in Table VIII, page 30.

Major Structural Members

Figure 60 shows the location of the major structural bearings in the four-point hoist.

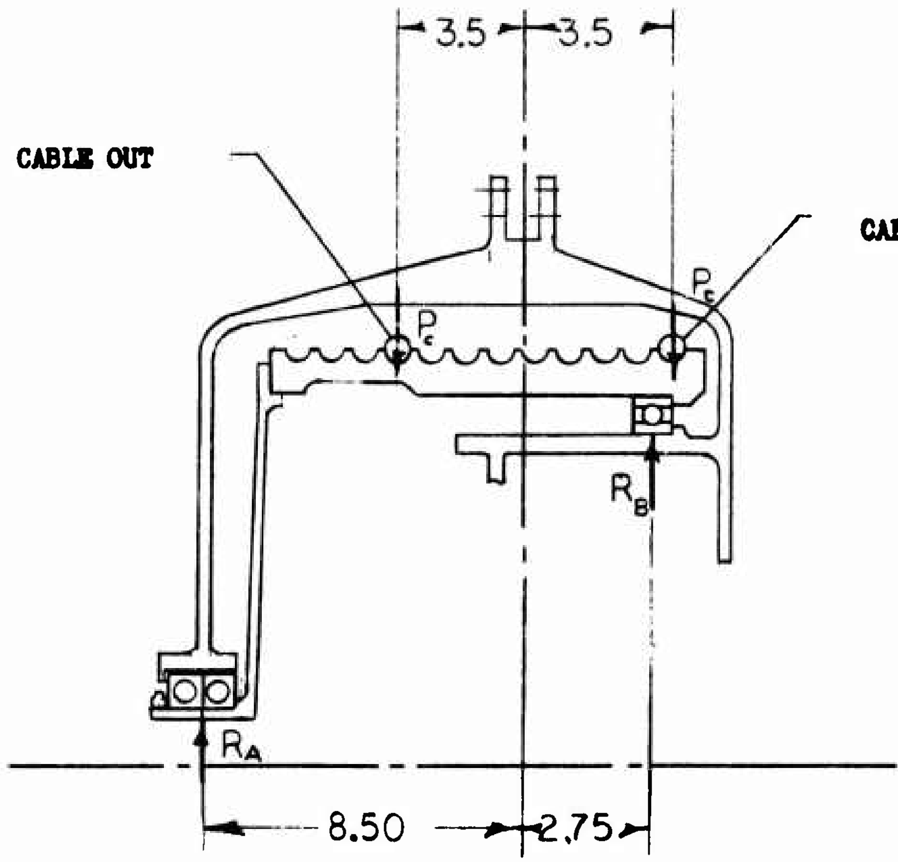


Figure 60. Major Structural Members, Four-Point Hoist.

Cable loads are transmitted to the side plates as tensile loads. No thrust can be transmitted because of the zero-moment mounting.

From a consideration of static equilibrium,

$$\left.\begin{aligned} R_A &= .0667P_c \\ R_B &= 1.0667P_c \end{aligned}\right\} \text{Cable In} \qquad \begin{aligned} (91) \\ (92) \end{aligned}$$

$$\left.\begin{aligned} R_A &= .5556P_c \\ R_B &= .4444P_c \end{aligned}\right\} \text{Cable Out} \qquad \begin{aligned} (93) \\ (94) \end{aligned}$$

The minimum cross section on the side of bearing B is shown in Figure 61 .

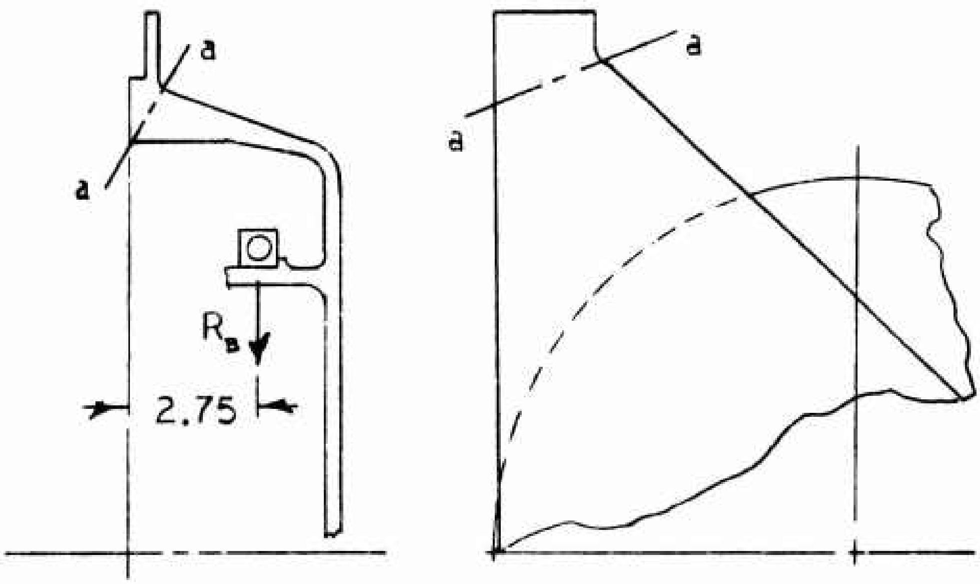


Figure 61. Side Plate, Four-Point Hoist.

At section a-a,

$$\text{base} = 6.5$$

$$\text{height} = 1.50$$

$$Z = \frac{bh^2}{6}$$

$$Z = \frac{6.5\,(1.50)^2}{6}$$

$$Z = 2.438 \tag{95}$$

$$M_b = 2.75\ R_B \tag{96}$$

for P_c = 48,500 lb (ultimate load)

$$M_b = 2.75\ (1.0667)\ (48{,}500)\ (\text{cable in})$$

$$M_b = 142{,}300\ \text{in.-lb}$$

From equation (6)

$$f_b = \frac{142{,}300}{2.438}$$

$$f_b = 58{,}370\ \text{psi}$$

Level Wind Mechanism

The level wind mechanism in the four-point hoist has axial forces induced due to the angle of the cable when it is in its limit positions. This condition is illustrated in Figure 62.

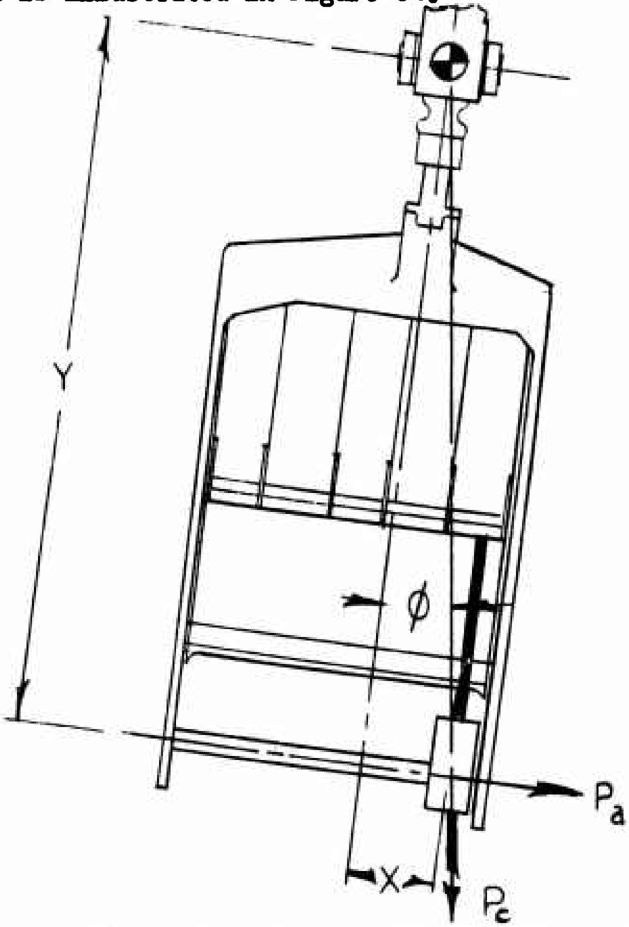


Figure 62. Induced Axial Loads in Level Wind Ball Screw, Four-Point Hoist.

From Figure 62,

$$\tan \phi = \frac{X}{Y} \tag{97}$$

$$P_a = P_c \sin \phi \tag{98}$$

For small angles $\sin \phi \approx \tan \phi$,

$$P_a = P_c \frac{X}{Y} \tag{99}$$

X = 3.5 in.

Y = 35.35 in.

$$P_a = P_c \frac{3.5}{35.35}$$

$$P_a = .099\ P_c$$

$$P_{a\ normal} = .099\ (11{,}550)$$

$$P_{a\ normal} = 1140 \text{ lb}$$

$$P_{a\ ult} = .099\ (48{,}500)$$

$$P_{a\ ult} = 4800 \text{ lb}$$

Ball screw data

Ball circle diameter	= 1.5	
Ball diameter	= 5/32	
Lead of thread	= 1/4	
Turns of balls	= 5	
Operating load	= 5240 lb	(Reference 11)
Max static load	= 31,500 lb	(Reference 11)

The normal load is less than the operating load for 1 million inches of travel, as shown in Reference 11. One million inches of travel in the four-point hoist corresponds to 71,430 cycles of operation (3830 hours).

The ratio from the drum to the ball screw is given by equation (49).

$$RR = \frac{1/4}{7/8}$$

$$RR = \frac{2}{7}$$

Hence the ball screw must turn 3.5 times as fast as the drum to keep the bellmouth of the level wind in the same relative axial position as the cable takeoff point on the drum.

If we assume that the ball screw shaft is simply supported, the critical buckling load can be found from equation (52).

$$L = 15$$

$$d_e = 1.4 \text{ (effective)}$$

$$d_i = .90$$

$$I = .1564 \text{ in.}^4$$

$$P = \frac{\pi^2 \, 30 \times 10^6 \, (.1564)}{(15)^2}$$

$$P = 206{,}000\text{-lb screw buckling load}$$

The torque required to turn the screw against the 1140-pound normal load is given by equation (53).

$$\eta = .90 \quad \text{(Reference 8)}$$

$$T_{normal} = \frac{1140 \, (.25)}{2 \, \pi \, (.90)}$$

$$T_{normal} = 50.4 \text{ in.-lb}$$

The scrub roll is a rubber coated roller whose pitch line velocity is 9.8 pct greater than the cable pitch line velocity when the cable is reeling out. A one-way clutch disengages the roller when the cable is reeling in. An adjustable pulley, which rides with the bellmouth and ball screw, is used to load the cable against the scrub roll rubs on the cable, keeping a slight tension in it to keep the cable tracking in the drum grooves when the hook is unloaded.

Weston Brake

The Weston brake in the four-point hoist is similar in design to the Weston brake in the single-point hoist. The lining and screw data are given below:

$$d_o = 5.25$$

$$d_i = 3.25$$

$$n = 6 \text{ friction surfaces}$$

$$\mu = .07$$

$$T = 1130 \text{ in.-lb} \quad (11{,}550\text{-lb cable load})$$

From equation (56),

$$P_a = \frac{8 \, (1130)}{\pi \, (.07)(3.25)(5.25^2 - 3.25^2) \, 6}$$

$$P_a = 124 \text{ psi}$$

Screw data:

$2\frac{1}{4} - 3$ triple thread

lead $= 1 = .3333$

$d_m = 1.917$

The lead angle, α, is calculated using equation (57).

$$\alpha = \arc \tan \frac{3\ (.33333)}{1.917}$$

$$\alpha = \arc \tan\ .16603$$

$$\alpha = 9^\circ 26'$$

This value meets the requirement for proper operation as given by equation (58).

Heat Rise in Weston Brake

When the Weston brake is used to lower the load, all the energy of the load must be absorbed by the brake. This energy is given by the work done per revolution times the number of revolutions required to lower the load. For a 11,550-pound load being lowered 50 feet at 31.06 fpm, the time to lower is calculated using equation (59).

$$t = \frac{50}{31.06} = 1.61 \text{ min}$$

Heat generated is given by equations (60) and (61).

$$H = \frac{2\ \pi\ (94.2)(608.8)(1.61)}{778}$$

$$H = 746 \text{ Btu}$$

It has been conservatively assumed that 50 pounds of metal and 10 pounds of oil which surround the Weston brake are affected by heat when lowering the load. The temperature rise after one lowering operation is calculated using equation (62).

$$\Delta t = \frac{746}{.12\ (50) + .55\ (10)}$$

$$\Delta t = 65^\circ F$$

Free Reeling Emergency Release

As in the single-point hoist, an emergency free reel release has been provided. This unit is a clutch whose plates are axially loaded by Belleville washers. A hydraulic piston compresses the washers to release the clutch plates. The clutch is designed to carry the ultimate cable load with a factor of safety of 1.25.

$$T_{design} = \frac{P\ d_m}{2\ RR} \times F.\ S.$$

$$T_{design} = \frac{48,500\ (22.19)}{2\ (113.86)} \times 1.25$$

$$T_{design} = 5,910 \text{ in.-lb} \qquad (100)$$

$$d_o = 3.00$$

$$d_i = 2.00$$

$$n = 14 \text{ friction surfaces}$$

$$\mu = .25 \text{ (static)}$$

Using equation (56), we obtain

$$P_a = \frac{8\ (5910)}{\pi\ (.25)(2.00)(3.00^2 - 2.00^2)\ 14}$$

$$P_a = 434 \text{ psi}$$

The axial clutch load necessary to produce this pressure is given by equation (64).

$$F_a = \frac{\pi\ (434)(2.00)}{2}\ (3.00 - 2.00)$$

$$F_a = 1360\text{-lb} \text{ axial load required by Belleville washers}$$

The Belleville washers design for the four-point hoist is similar to the Belleville washer design for the single-point hoist. The same stacking arrangement is used: two sets in series of three washers in parallel for each set. Using the nomenclature of Figure 57, page 179, the data and results are as follows:

$R = 1.88$
$r = 1.06$
$t = .069$
$h = .103$
$C = 1.57$
$C_1 = 1.50$
$\delta = .050$
$\Delta = .100$

Clutch plate clearance = .106

With these values substituted in equation (65) the force required to flatten the springs is 1450 pounds.

Drum Speed After Free Fall

By substituting appropriate values in equation (76), the curves shown by Figure 63 have been plotted to show the ratio of rpm before and after fall versus distance of fall for various cable loads.

As was the case in the single-point hoist, in the four-point hoist the inertia has little effect over that of a free falling weight, when the weight is large.

The free-reeling clutch shaft is normally rotating at 608.8 rpm. After free fall of 50 feet with a cable load of 11,550 pounds, the free reeling clutch shaft will be rotating at 65,780 rpm. At this high speed, rotational stresses will be induced into the outer clutch plate holder (largest member on shaft). For this part,

$d_o = 3.90$

$d_i = 3.25$

$\rho = .283 \text{ lb/in.}^3$

$\nu = .3$

Using equation (80), we obtain

$$\omega = 65780 \frac{\pi}{30} = 6888 \text{ rad/sec}$$

Solving equation (78), we obtain

$$f_{r\,max} = \frac{(3 + .3)}{32} \frac{(.283)(6888)^2}{386} (3.90 - 3.25)^2$$

$$f_{r\,max} = 1520 \text{ psi}$$

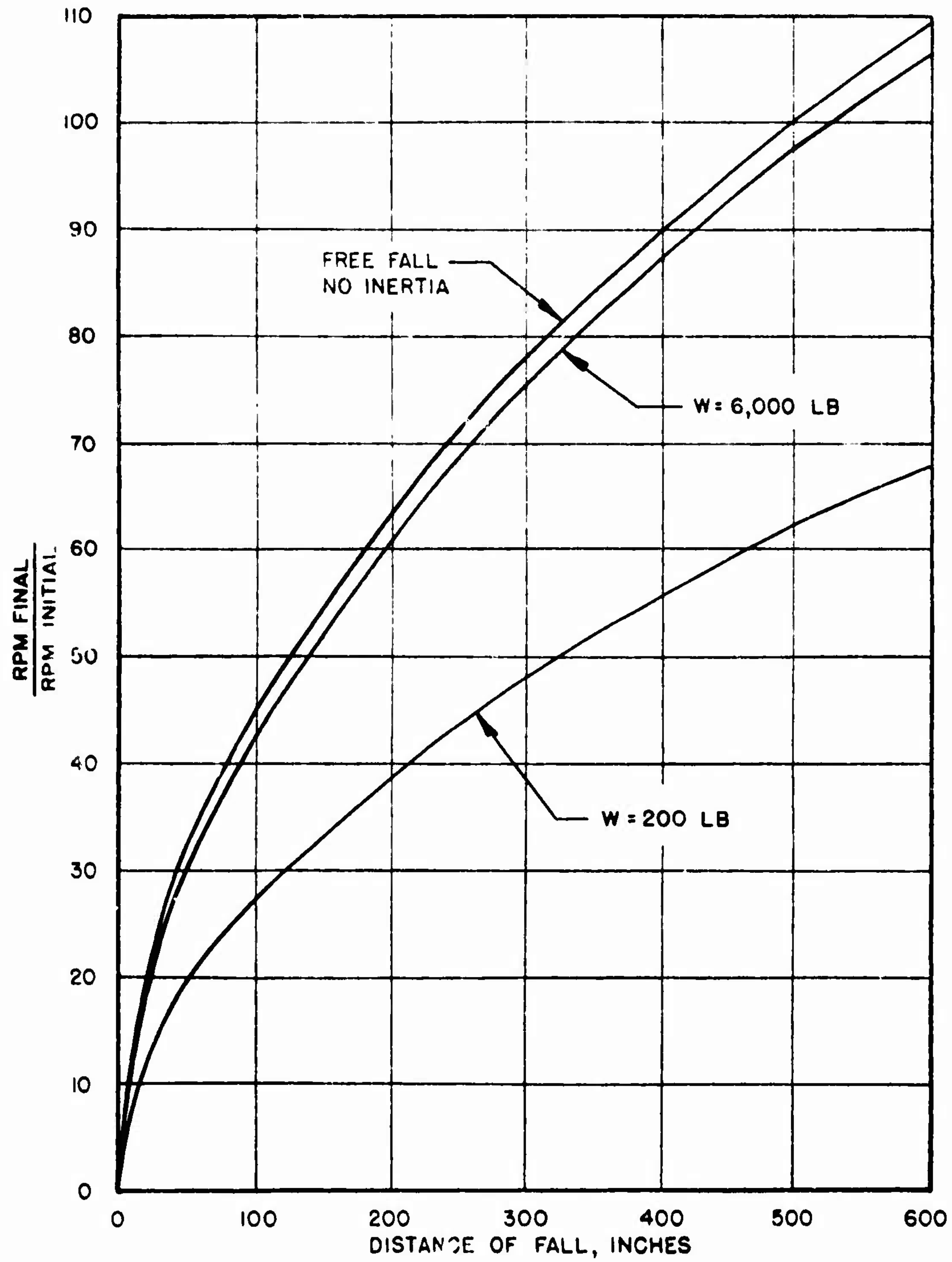


Figure 63. Load vs Free Fall Velocity, Four-Point Hoist.

Solving equation (79), we obtain

$$f_{t\ max} = \frac{.283\ (6888)^2}{16\ (386)} \left[(3 + .3)(3.90)^2 + (1 - .3)(3.25)^2\right]$$

$$f_{t\ max} = 125{,}200 \text{ psi}$$

The free reeling clutch output plate is made from AMS 5000 steel whose ultimate tensile strength is 200,000 psi and whose yield strength is 176,000 psi. The maximum tensile stress produced in this part by high-speed rotation is therefore below the yield strength.

Gear Design

The bending and compressive stresses in the gear teeth of the four-point hoist are found by methods similar to those used on page 185 for the single-point hoist. The level wind drive for the four-point hoist is a chain and sprocket arrangement and is covered in another section. Table XXXII summarizes the bending and compressive stresses for all the spur gear teeth on the four-point hoist.

Planetary Plate Design

The planetary carrier plates of the third and fourth stages of gearing in the four-point hoist are subjected to steady bending stresses. 6AL-4V titanium is used because it is lighter than steel plates. The plates may be designed for maximum slope or maximum stress. The allowable stress and slope is given below:

$$f_{b\ allow} = 40{,}000 \text{ psi}$$

$$\phi_{allow} = .001 \text{ inch per inch}$$

Plate stress is given by

$$f_b = \frac{12\ T\ L}{n\ d_s\ l\ (d_o - d_i - 1.2\ d)} \left[\frac{g + t}{t^2}\right] \qquad (101)$$

where

T = sun gear torque

$$L = \text{geometry factor} = \frac{1 - .8d}{2} \qquad (102)$$

d_s = sun gear diameter

TABLE XXXII

GEAR SUMMARY - FOUR-POINT HOIST

	N Number of Teeth	Pitch	d Diameter	F Face Width	X Tooth Form Factor	K_t Stress Conc. Factor	Weight Tang. tooth Load at 11,550 lb Cable Load	f_b Bending Stress	f_c Hertz Stress
Input Spur Pinion	29	16	1.8125	.58	.0594	1.38	286	17,180	127,100
Input Spur Idler	29	16	1.8125	.58	.0594	1.38	256	17,180	127,100
Input Spur Gear	131	16	8.1875	.50	.0803	1.50	286	16,030	99,340
2nd Stage Spur Pinion	22	16	1.375	1.90	.0542	1.36	1700	33,680	149,700
2nd Stage Spur Gear	138	16	8.625	1.90	.0804	1.50	1700	25,040	149,700
3rd Stage Plan. Sun Gear	31	12	2.583	1.22	.0808	1.25	1420	27,010	149,900
3rd Stage Plan. Pinion	45	12	3.750	1.31	.0902	1.26	1420	22,710	149,900
3rd Stage Plan. Ring Gear	121	12	10.083	1.18	.1472	1.36	1420	16,680	77,370
4th Stage Plan. Sun Gear	47	10	4.700	1.55	.1095	1.20	2550	27,040	149,900
4th Stage Plan. Pinion	40	10	4.000	1.68	.1046	1.19	2550	25,900	149,900
4th Stage Plan. Ring Gear	127	10	12.700	1.50	.1759	1.34	2550	19,430	92,990

$$l = (d_s + d_p) \sin \frac{\pi}{n} \tag{103}$$

n = number of pinions

d_p = pinion diameter

d = O.D. of inner race of pinion bearing

g = distance between plates

t = thickness of single plate

d_o = outside diameter

d_i = inside diameter

Plate slope is given by

$$\phi = \frac{16\ T\ L^3}{n\ d_s\ E\ l^2\ (d_o - d_i)} \left[\frac{(g + t)}{t^3} \right] \tag{104}$$

where the symbols are the same as those used for the stress formula.

For the 3rd stage planetary,

g = 1.50 in.

t = 7330 in.-lb (11,550-lb cable load)

d_s = 2.583 in.

d_p = 3.75 in.

n = 4 pinions

d = 2.31 in.

l = 4.478 in.

L = 1.315 in.

d_o = 9.84 in.

d_i = 2.75 in.

t = .28 in.

E = 16×10^6 psi

Substituting in equation (101), we obtain

$$f_b = \frac{(12)(7330)(1.315)}{(4)(2.583)(4.478)(9.84 - 2.75 - 1.2 \times 2.31)} \quad \frac{(1.50 + 28)}{(.28)^2}$$

$$f_b = 13{,}140 \text{ psi}$$

Substituting in equation (104) we obtain

$$\phi = \frac{(16)(7330)(1.315)^3}{(4)(2.583)(16 \times 10^6)(4.478)^2(9.84 - 2.75)} \quad \frac{(1.50 + 28)}{(.28)^3}$$

$$\phi = .00092 \text{ inch per inch}$$

For the 4th stage planetary,

g = 1.83 in.

T = 35,960 in.-lb (11,550-lb cable load)

d_s = 4.700 in.

d_p = 4.000 in.

n = 6 pinions

d = 2.18 in.

l = 4.350 in.

L = 1.303 in.

d_o = 12.375 in.

d_i = 5.000 in.

t = .36 in.

E = 16×10^6 psi

Substituting in equation (101), we obtain

$$f_b = \frac{(12)(35,960)(1.303)}{(6)(4.70)(4.35)(12.375 - 5.0 - 1.2 \times 2.18)} \quad \frac{(1.83 + .36)}{(.36)^2}$$

$$f_b = 16{,}280 \text{ psi}$$

Substituting in equation (104), we obtain

$$\phi = \frac{(16)(35,960)(1.303)^3}{(6)(4.70)(16 \times 10^6)(4.350)^2 \, (12.375 - 5.00)} \quad \frac{(1.83 + .36)}{(.36)^3}$$

ϕ = .00095 inch per inch

Level Wind Drive Chain

The level wind drive chain is designed for a normal torque of 50.4 inch-pounds. The chain and sprocket data are given below:

Number of Teeth in Drive Sprocket	=	63
Number of Teeth in Driven Sprocket	=	18
Diameter of Drive Sprocket	=	7.560
Diameter of Driven Sprocket	=	2.160
American Standard Chain #35		
Chain Pitch	=	.375
Average Tensile Strength	=	2100 lb
Rated HP for 17-Tooth Sprocket	=	.061
Correction Factor for 18-Tooth Sprocket	=	1.05
RPM of Driven Sprocket	=	18.7

$$HP = \frac{T \times rpm}{63,025}$$

$$HP = \frac{50.4\ (18.7)}{63,025}$$

$$HP = .015 \qquad (105)$$

This is well below the allowable HP of .061 from Reference 12 for 15,000 hours of chain life. The maximum tensile load in the chain is given by

$$P_{max} = \frac{2\ T\ normal}{d} \left(\frac{P_{cable\ ult}}{P_{cable\ normal}}\right)$$

$$P_{max} = \frac{2\ (50.4)}{2.160} \frac{(48,500)}{(11,550)}$$

$$P_{max} = 196 \text{ lb} \qquad (106)$$

This is well below the average tensile strength given in Reference 12.

Bearing Design

Shaft bearing reaction caused by gear loads may be found by well-known methods, as shown in Reference 5. All bearing loads, ratings, and lives are tabulated in Table XXXIV for normal and static ultimate conditions.

Drive System Shaft Stresses

The drive train shafting of the four-point hoist may be subjected to bending stresses, torsional stresses, or a combination of both. In bending,

the hoist shafts are critical under normal load fatigue conditions, since the endurance limit multiplied by the ultimate load factor is less than the ultimate tensile strength of the material. In torsion, however, the shafts are critical under ultimate load conditions. The four-point hoist shafting has been analyzed by the methods of Reference 3, and the results are given in Table XXXIII.

TABLE XXXIII
CRITICAL SECTION SHAFT STRESSES - FOUR-POINT HOIST

Location	Critical Section	Type of Critical Stress	Stress (psi)	M.S.
Input Pinion Shaft	Output Pinion Brg. Rad.	Bending	5,770	2.47
Input Idler Shaft	Bearing Radius	Bending	8,030	1.49
Input Gear Shaft	Spline Undercut	Bending	1,100	17.18
Free Reel Clutch Input	Torque Shaft	Torsion	21,070	3.13
Free Reel Clutch Output	Torque Shaft	Torsion	57,610	.51
2nd Stage Pinion Shaft	Bearing Radius	Bending	8,930	1.24
2nd Stage Gear Shaft	Webb Connection	Bending	12,280	.63

Note: Torsional stresses are critical under ultimate load conditions.

Cable Breaking Strength

For the four-point hoist the G factor is 2.8 and the factor of safety for ultimate load conditions is 1.5. Using equation (85) to determine the required cable breaking strength, we obtain

$$P_{ult} = 11{,}550\,(2.8)(1.5) = 48{,}500 \text{ lb}$$

The cable to be used has 342 wires (18 x 19 construction) that are .035-in. diameter. Solving equation (88) to determine cable area, we obtain

$$A = 18 \times 19 \times \frac{\pi}{4} \times .035^2 = .329 \text{ in.}^2$$

Solving equation (87) to obtain cable breaking strength, we obtain

$$P = 250{,}000\,(.71)(.329) = 58{,}400 \text{ lb}$$

TABLE XXXIV
SUMMARY OF BEARING LIVES AND LOADS-
FOUR-POINT HOIST

Location	Special Load Condition	RPM	Brg. Loa Cable (P = 32 Thrust
Input Pinion, Gear End *		2750	-
Input Pinion, Outboard End *		2750	-
Input Idler, Outside End *		2750	-
Input Idler, Inside End *		2750	-
Input Gear, Outside End *		608.8	-
Input Gear, Weston Brake End *		608.8	-
Free Reel Clutch Main Timken		0	1360
Free Reel Clutch Piston Isolator**	Free Reel Piston On	608.8	1450
Free Reel Clutch Washer Preload**	Free Reel Piston On	608.8	1450
2nd Stage Pinion, Input End (Roller)		608.8	0
2nd Stage Pinion, Outboard End (Roller)		608.8	0
2nd Stage Gear, Input End		97.1	0
2nd Stage Gear, Planetary End		97.1	0
3rd Stage Planetary, Pinion Bearing		53.2	0
4th Stage Planetary, Pinion Roller Brg.		17.0	0
Level Wind Ball Screw Thrust Bearing	Cable Max. In or Out	18.7	3200
Main Drum Support Bearing (Brg. B Fig. 60)	Cable In	5.3	0
Secondary Drum Support Brg. (Brg. A Fig.60)	Cable Out	5.3	0

Note: Any bearings not shown carry no load (or negligible load
*Static loads not felt from input to Westom brake.
**Free reel bearings loaded only when free reel piston act:

A.

TABLE XXXIV
SUMMARY OF BEARING LIVES AND LOADS-
FOUR-POINT HOIST

Special Load Condition	RPM	Brg. Load @ Limit Cable Load (P = 32,300 lb) Thrust	Radial	Static Cap. 1.25 Co (3 Co for Zero RPM)	Brg. Load @ Normal Cable Load (P - 11,550 lb) Thrust	Radial	Dynamic Capacity	Life (Hours)
	2750	-	-	-	0	495	1,740	263
	2750	-	-	-	0	165	1,740	7,110
	2750	-	-	-	0	286	965	233
	2750	-	-	-	0	286	965	233
	608.8	-	-	-	0	152	5,090	1,030,000
	608.8	-	-	-	0	178	5,280	714,400
	0	1360	0	25,700	1360	0	-	
Free Reel Piston On	608.8	1450	0	14,900	1450	[illegible]	-	
Free Reel Piston On	608.8	1450	0	11,400	1450	0	-	
	608.8	0	2570	4,150	0	920	5,420	9,860
	608.8	0	2570	4,150	0	920	5,420	9,860
	97.1	0	3240	4,780	0	1,160	4,890	12,860
	97.1	0	1900	2,570	0	680	2,300	6,640
	53.2	0	7940	8,000	0	2,840	5,520	1,330
	17.0	0	14,260	17,850	0	5,100	19,100	43,480
Cable Max. In or Out	18.7	3200	170	7,650	1140	60	5,270	19,290
Cable In	5.3	0	34,450	53,400	0	12,320	14,290	4,860
Cable Out	5.3	0	17,950	23,600	0	6,420	16,200	50,030

t shown carry no load (or negligible load).
t felt from input to Weston brake.
ngs loaded only when free reel piston activated.

B.

WEIGHT ANALYSIS

A weight analysis based on the preliminary design (layout) drawings of Appendix III has been made of all the components comprising the 40,000-pound-capacity external cargo handling system. A detailed weight breakdown for the mechanically driven single-point hoist with a usable cable length of 100 feet and the hydraulically powered four-point hoists is presented in Table XXXV. For comparison with the weight estimates of Phase I, the weight for an 80-foot single-point hoist is included in parentheses.

The calculated weight of the complete single- plus four-point system is 4974 pounds, including controls wiring and aircraft supporting structure peculiar to the hoist system. Of this, approximately 2200 pounds (single-point hoist and input drive shaft) is readily removable when missions requiring minimum aircraft empty weight are to be undertaken utilizing four-point suspension. Similarly, if only the single-point hoist is to be used, the four-point hoists can be removed, providing a weight reduction of 2236 pounds. The weight of the cargo handling system chargeable to aircraft empty weight is then:

Single-Point Mission (four-point removed) = 2738 lb

Four-Point Mission (single-point removed) = 2704 lb

TABLE XXXV
WEIGHT SUMMARY-
40,000-POUND EXTERNAL CARGO HANDLING SYSTEM

Item	Weight (lb) Component	Assembly	System
Single-Point Hoist System			2406 (2133)*
Single-Point Hoist		2270 (1960)*	
Drum and Bearings	353		
Gearing and Housings	589		
Supports and Bearings	180		
Level Wind Assembly	257		
Potentiometer, Cable Cutters	14		
Anti-Backlash Cover	58		
Lubricating Oil (4 gal)	28		
Free Reel Unit and Controls	71		
Cable	440		
Hook, Swivel, & Slip Ring	150		
Decoupler (Isolator)	130		
Drive Train		120 (173)*	
Clutch-Reverser	46		
Upper Angle Gearbox	21		
Lower Angle Gearbox	25		
Shafting and Bearings	28		
Control Unit, Display Wiring, Etc.		16 (0)*	
Four-Point Hoist System			2568
Four-Point Hoist		559	
Drum and Bearings	89		
Gearing and Housings	184		
Level Wind and Supports	48		
Potentiometers, Potentiometer Clutches, Switches, Cable Cutters	13		
Anti-Backlash Cover	16		
Lubricating Oil (2 gal)	14		
Free Reel Unit	19		
Cable	76		
Hook, Swivel, & Slip Ring	52		
Isolator	38		
Hydraulic Motor	10		

TABLE XXXV (continued)

Item	Weight (lb) Component	Assembly	System
Hydraulic Subsystem**		111	
Hydraulic Pump	20		
Lines, Fittings, Fluid	90		
Filters	18		
Flow Divider Valves	12		
Relief Valve	2		
Shutoff Valves	4		
Free Reel Control		36	
Display, Control Boxes, & Wiring		35	
Structure (2 Davits)		150	

Single- Plus Four-Point Cargo Handling System 4974

*Weights in parentheses are those estimated in Phase I for hoist with 80 feet of cable.

**Hydraulic pump, filters, relief valve, and a portion of the lines and fittings will also be used for engine starting system. Weight chargeable to engine starting is 35 pounds.

MAINTAINABILITY AND RELIABILITY

INTRODUCTION

A comprehensive maintainability and reliability study was made of the heavy lift cargo handling system. This study was based on the system description of pages 134 through 156 and the drawings of Appendix III. The reliability and maintainability characteristics indicate that a combined system reliability of .969 can be expected, with an estimated .0718 maintenance man-hours per flight hour. A failure mode and effect analysis was also completed.

Considerable effort was also expended in a study of the safety aspects of the system; primary emphasis was placed on capability of jettisoning a load carried on the four-point suspension system. A reliability block diagram of the cable cutter system was prepared. This diagram shows that the least redundancy occurs at the explosive charge and cutter and indicates that primary emphasis should be placed on adequate testing of these components to achieve maximum safety.

RELIABILITY AND MAINTAINABILITY CHARACTERISTICS

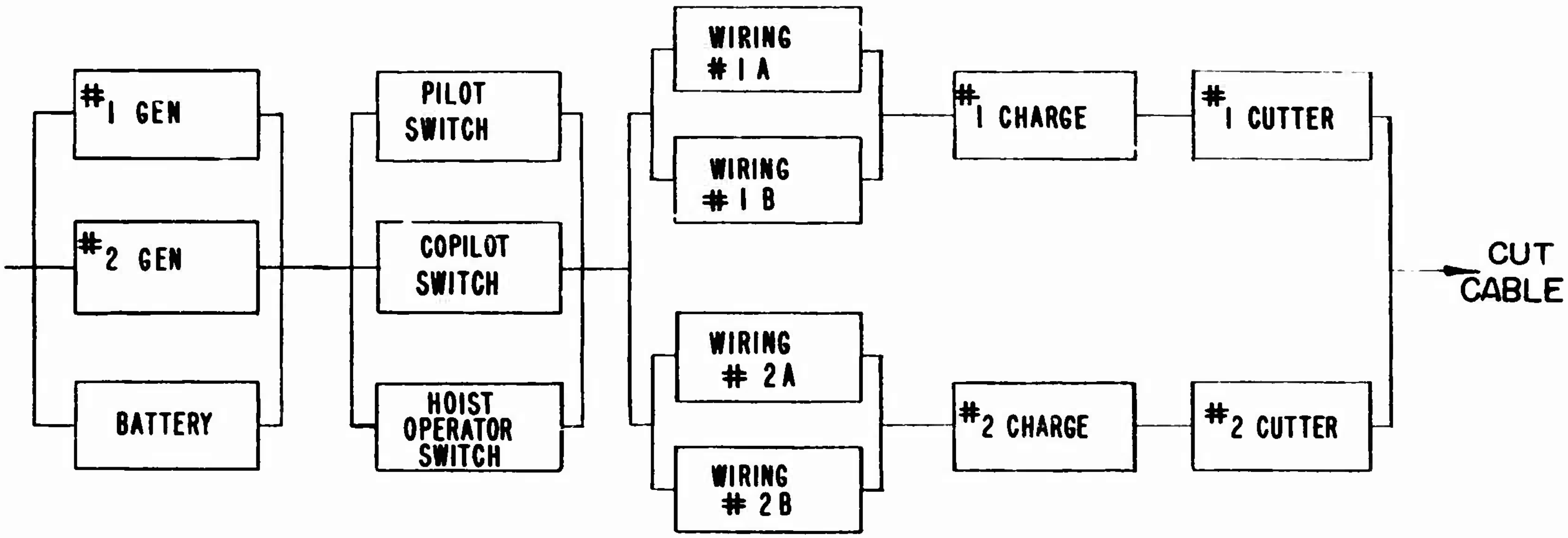
The selection of a mechanical drive for the single-point hoist and a hydraulic drive for the four-point hoists greatly increases the reliability and maintainability of the cargo handling system. Detail design, based on field experience with the cargo handling system on the CH-54A,has served to maintain these qualities, as shown in Table XXXVI, page 220.

FAILURE MODE AND EFFECT ANALYSIS

The results of a failure mode and effect analysis are summarized in Table XXXVII, page 222. Failure modes considered were only those that the reliability analysis indicated were the most probable. In all cases, the design is such that these failure modes are minimized, or features are included in the basic design of the system, to avoid detrimental effects on the mission.

SAFETY CONSIDERATIONS

Considerable time has been spent studying the safety aspects of the cargo handling systems. The single most critical consideration is the jettisoning of a four-point load. Should one of the attachment points fail to be released, the results could be catastrophic. Because of this, tandem-dual cable cutters at each hoist are used. The wiring is redundant to each cable cutter, and further, the redundant wiring is routed through the airframe in such a way as to reduce the vulnerability of the cutter charge ignition circuit significantly. This redundancy is required to ensure the proper level of safety.



Probability of success $= R_c = (1-Q_1Q_2Q_3)(1-Q_4Q_5Q_6)\left\{ 1-\left[1-(Q_7Q_8)R_{11}R_{12}\right]\left[1-(1-Q_9Q_{10})R_{13}R_{14}\right]\right\}$

Probability of success for four cables $= R_{4c} = R_c{}^4$

where
R = reliability

Q = unreliability

Figure 64. Reliability Block Diagram of Cable Cutter System.

TABLE XXXVI
RELIABILITY AND MAINTAINABILITY CHARACTERISTICS

	Single-Point System	Four-Point System	Combined System
Mission Reliability *	.995	.986	.9999**
System Reliability *	.985	.933	.969***
MTBUMA, Hours	32.5	7.2	15.8
Mean Time to Repair, Hours	.383	.364	.377***
Maintenance Burden, MMH/FH	.039	.148	.0718***
Inherent Availability, Percent	99	96	97.8***

*Based on a 30-minute mission and two complete cycles.

**Assumes complete redundancy of systems.

***Based on a 70-30 distribution of single- to four-point missions.

The ignition circuit is provided with a built-in test circuit which is used on the preflight check to ascertain that all circuits are functional.

Figure 64 is a reliability block diagram of the cable cutter system provided for each hoist. This diagram shows the redundancy employed in each of the systems. Note that the electrical source and the control switch are shared for each hoist.

In this diagram each parallel path provides a successful operation for that particular function. The diagram for the successful cutting of all the cable in the four-point system would show four of the diagrams of Figure 64 in series.

From the diagram it can be seen that the least redundancy is at the charge and cutter. Therefore it is recommended that adequate testing of these components be undertaken to ascertain with a high degree of confidence that their reliability is consonant with the overall reliability desired of this system.

INSTALLATION AND REMOVAL

Single-Point Hoist

Removal is accomplished simply by disconnecting the input shaft, loosening

6 bolts, and disconnecting the electrical lines at a quick disconnect fitting. The hoist is lowered from the aircraft by means of a special support equipment bridle and cable attached to the lifting points on the hoist support structure. The cable passes through airframe mounted pulleys and can be attached to a truck or wheeled vehicle or vehicle winch to lower the hoist to the ground.

Four-point hoist removal is accomplished by loosening 4 bolts and disconnecting with hydraulic and electrical quick disconnects. The four-point hoists are lowered by means of the same cable bridle system, which is supported over a similar pulley arrangement, as that utilized for the single-point hoist.

TABLE XXXVII
FAILURE MODE AND EFFECT ANALYSIS

	Component	Failure Mode	Effect	Special Design Features
Single-Point Hoist System	Hoist:			
	Bellmouth	Wear	None unless excessive to point of fraying cable	Field replaceable liner
	Scrub Roller	Wear	Excessive wear increases risk of cable fouling on drum	Adjustable and field replaceable
	Cable	Fraying	None unless allowed to progress to point of weakening cable	Bellmouth liner softer than cable; single wrap drum; proper design criteria for appropriate radii; field replaceable
		Broken conductor	Loss of corresponding electrical circuit	Spare conductor in cable
	Slip Ring Assembly	Failure to conduct or shorted out	Dependent on circuit but not critical	Sealed assembly to prevent environmental contamination; field replaceable
	Potentiometer	Improper output	Improper indication of cable length	Sealed unit, conservative design; limit switch prevents excessive reel-in or pay-out of cable; field replaceable
	Clutch Reverser	Leakage	None unless allowed to progress to point of losing most of oil	Motion on seal only when actually operating

TABLE XXXVII continued

Component	Failure Mode	Effect	Special Design Features
Clutches	Excessive wear	Clutch slippage - progressive from little effect to inability to raise and/or lower load; not critical	Conservative design; oil bath, normally unclutched
Angle Gearbox Seal	Leakage	None unless allowed to progress enough to deplete oil	Motion on seals only when operating hoist
Shafting Coupling	Fatigue crack of laminate	None unless allowed to progress to complete failure	Conservative design; hoist load brake holds load if coupling fails
Hook Assy. Micro-switch	Out of adjustment	Improper or no indication of hook status	
Solenoid	Open circuit	Loss of automatic opening and closing of hook	Conservative design; manual backup mechanism provided
Swivel Assy. Slip Ring	Failure to conduct or shorted out	Loss of circuit(s) effected; not critical	Completely sealed unit to prevent environmental contamination

TABLE XXXVII continued

	Component	Failure Mode	Effect	Special Design Features
Four-Point Hoist System	Hoist Assy.*			
	Hook Assy.*			
	Swivel Assy.*			
	Hydraulic Motor	Leakage	Performance degradation	Control system will compensate for performance degradation
	Hydraulic Pump	Leakage	Performance degradation	Pump can operate at reduced power output
	Flow Divider	Servo out of track	Loss of proper automatic control	Individual beeping of hoists may compensate for faulty automatic controls; cables may be reeled out to desired length & locked into position
	Potentiometer	Open circuit	Loss of proper automatic control	See above
	Amplifier	Output out of tolerance	Loss of proper automatic control	
	Clutch	Fails to deliver torque	Improper synchronization	Magnetic particle clutch provides trouble-free service because of no contact between the driving and driven elements

*See single-point hoist.

GROWTH POTENTIAL

Both the mechanically driven single-point hoist system and the hydraulically driven four-point hoist system have potential for increased capacity. Growth to 50,000 pounds for these systems can be accomplished in incremental stages of modification and payload increase. The following paragraphs summarize the stages of improvement or "break-points", the modifications necessary, and the estimated weight increase.

SINGLE-POINT HOIST SYSTEM

Design Data:

Capacity (pounds)	40,000
Ultimate Load Factor	3.75
Usable Cable Length (feet)	100
Cable Speed (ft/min)	60
Power Requirements (HP)	94.2

Stage 1 Growth — Capacity = 45,000 lb

The first "break-point" in the single-point hoist system growth is in the ultimate (breaking) strength of 171,600 pounds for the single-point hoist cable (see page 187). For an ultimate load factor of 3.75, the maximum permissible cable load is

$$P_{cable} = \frac{171,600}{3.75} = 45,800 \text{ lb}$$

The power required at 45,000-lb capacity is 106 HP. At this load and power level, the only modifications necessary in the single-point hoist system are as follows:

1. Drum (p. 30)	Increase thickness from 1.44 to 1.65 inches.
2. Planet Pinion Bearings (p. 188)	Increase bearing size from 114 series to 212 series for increased static capacity.
3. Level Wind Ballscrew Bearings (p. 189)	Increase taper roller bearing size for increased static capacity.

The estimated weight increase for the stage 1 growth is 62 pounds.

Stage 2 Growth — Capacity = 50,000 lb

To obtain a capacity of 50,000 pounds at the 3.75 load factor, a 1.52-inch-diameter 18 x 19 cable is required. Maintaining the same drum diam-

eter and width reduces the useful cable length to 90 feet. The power required is 118 HP.

The necessary modifications to the hoist are

1. Cable (p. 187)	New 18 x 19, 1.52 inch diameter.
2. Cable Drum (p. 30)	Increase thickness to 1.85 inches and change pitch to 1.625 inches.
3. Planet Pinion Bearings (p. 188)	Same as Stage 1 change.
4. Level Wind Ballscrew and Nut (p. 172)	Change pitch to .541 inch.
5. Level Wind Ballscrew Bearings (p. 189)	Same as Stage 1 change.

The estimated weight increase for the above modifications is 170 pounds.

FOUR-POINT HOIST SYSTEM

Design Data:

System Capacity (pounds)	40,000
Hoist Capacity (pounds)	11,550
Ultimate Load Factor	4.2
Usable Cable Length (feet)	50
Cable Speed (ft/min)	30
Power Requirements (HP)	84.8
Hydraulic Requirements	
Pressure (psi)	3500
Flow (fpm/hoist)	11.1

Stage 1 Growth

System Capacity = 48,000 lb
Hoist Capacity = 13,900 lb

As was the case with the single-point hoist system, four-point capacity growth is paced by the ultimate (breaking) strength of the cable. The .79-inch cable has a breaking strength of 58,400 pounds (see page 212), giving a dynamic capability of 13,900 pounds. The modifications to the system to achieve this capability are

1. Drum (p. 30)	Increase thickness from 0.81 to 0.97 inch.
2. 1st Stage Planet Pinion Bearings (p. 213)	Increase bearing size from 1911 to 110 series for increased static capacity.

3.	2nd Stage Planet Pinion Bearings (p. 213)	Increase bearing size from 110 to 208 series for increased static capacity.
4.	Planetary Carrier Plate (p. 207)	Increase plate thickness to .31 to reduce deflector.
5.	Hydraulic System	Increase pressure from 3500 to 4000.

The estimated weight increase for the above modifications is 25 pounds per hoist.

Stage 2 Growth

System Capacity = 50,000 lb
Hoist Capacity = 14,450 lb

To achieve a four-point hoist system capacity of 50,000 pounds at the 4.2 load factor, a cable with a breaking strength of 60,700 pounds is required; inasmuch as this is such a minor increase over the actual cable strength of 58,400 pounds, a small reduction in ultimate load factor is recommended. The load factor at 50,000 pounds is

$$\text{Ultimate Load Factor} = \frac{58,400}{14,450} = 4.03$$

In addition to increasing the hydraulic system pressure to 4000 psi (as is required for Stage 1), it will also be necessary to increase the hydraulic motor displacement by 12 pct by reboring the existing motor.

COMPONENT AND SYSTEM DEVELOPMENT PLAN

DISCUSSION

The following paragraphs outline the development effort suggested for the heavy lift cargo handling system of this report. For the endurance test phase of both the single- and four-point hoists, the construction of a special test facility that can be used for both the single- and four-point hoist systems will be required. No attempt has been made to provide a schedule for the fabrication of such a facility, although its estimated costs are presented on page 230.

While the plan presented is directed toward the systematic development of the full-size aircraft hardware, several areas are suitable and are recommended for development effort in advance. Among these are the investigation of synchronous operation of the four-point hoist system (where the hydroelectrical feedback system can be developed in model form) and the reliability of hook release of both the single- and four-point hoists.

TEST PROGRAM

The following test programs are recommended to develop and qualify the heavy lift external load handling system.

Mechanically Driven Single-Point Hoist

All components of drive train and single-point hoist will be utilized in all phases of the test program except the static load test.

1. Static Load Test:

   Conduct a static load test on single-point hoist to limit load (100,000 lb)

2. Endurance Test:

   Conduct a 3600-cycle endurance test at the following load spectrum:

Cycles	Distance (ft)	Load (lb)	Cable Position
800	100	40,000	vertical
2400	100	30,000	vertical
200	100	20,000	vertical
100	100	0	vertical
50	100	30,000	15° lateral

Cycles	Distance (ft)	Load (lb)	Cable Position
50	100	30,000	15° aft

3. Environmental Test:

Conduct an environmental test on the single-point hoist and drive train.

High Temperature Test
Low Temperature Test
Sand and Dust Test
Rain Test

Note: All tests to be in accordance with MIL-E-5272C

Hydraulically Driven Four-Point Hoists

1. Static Load Test:

Conduct a static load test on a four-point hoist to limit load (29,000 lb).

2. Endurance Test:

Conduct a 3600-cycle endurance test on a four-point hoist at the following load spectrum:

Cycles	Distance (ft)	Load (lb)
800	50	11,550
	13	11,550
2400	50	8,650
	13	8,650
200	50	5,800
	13	5,800
200	50	0
	13	0

3. Environmental Test:

Conduct an environmental test on the four-point hoist, control system, hydraulic pump, motor, and lines to the following requirements:

High Temperature Test
Low Temperature Test
Sand and Dust Test
Rain Test

Note: All tests to be in accordance with MIL-E-5272C

4. System Test:

Conduct a 200-cycle test to demonstrate synchronous operation of the four-point hoist system and the adequacy of the hydroelectrical feedback control system at varying load conditions.

Hook Swivel Assemblies

1. Qualification and Environmental Tests:

Conduct a 3600-cycle release test on both the single- and four-point hook assemblies.

Conduct separate environmental tests at the conditions outlined for hoists.

2. Functional Test:

Conduct a hook functional test in conjunction with respective hoist endurance test.

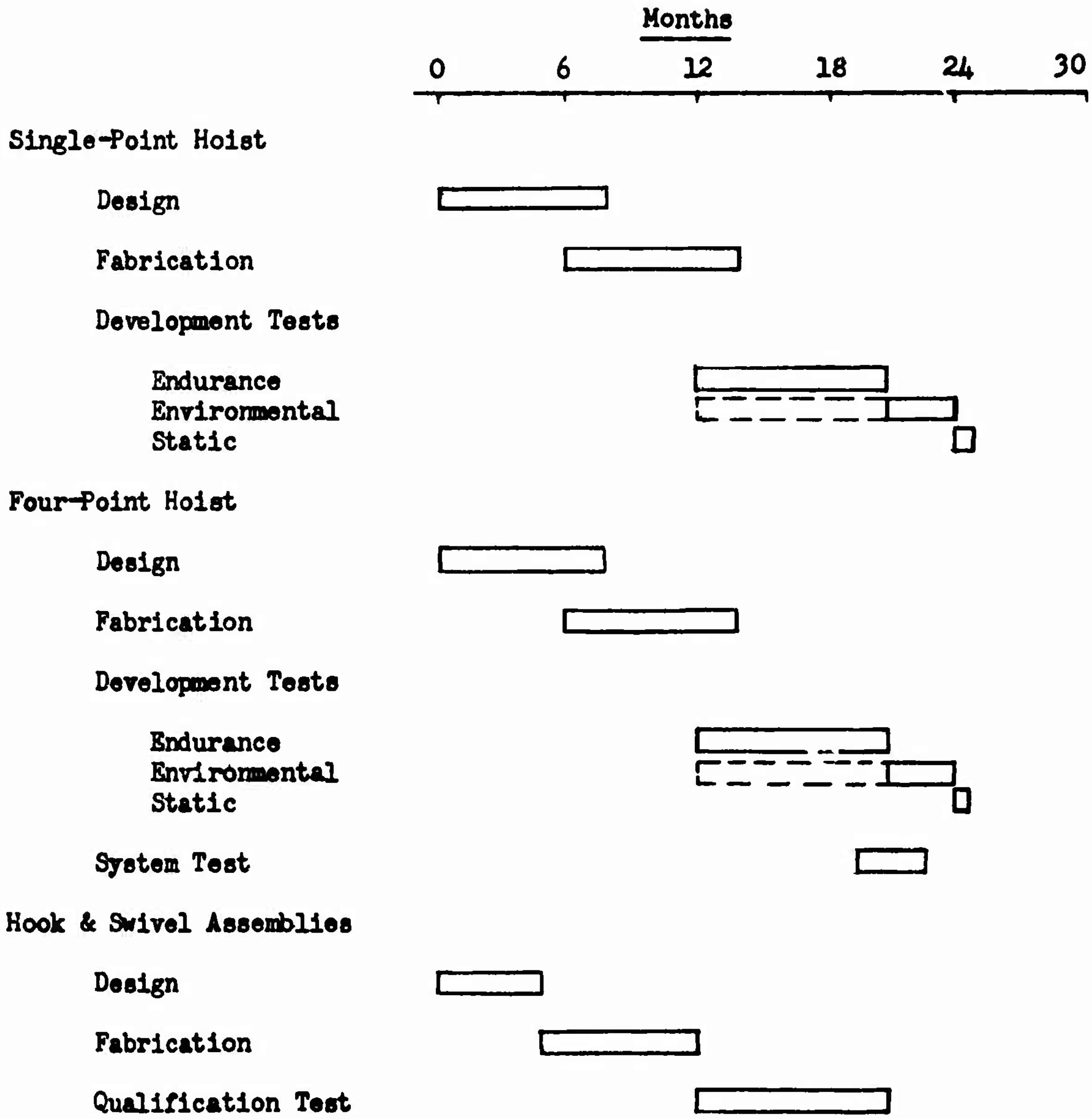
ESTIMATED DEVELOPMENT COSTS

Based on preliminary data available at this time, it is anticipated that the configuration recommended can be developed for the estimated prices set forth below:

	Single-Point	Four-Point
Engineering Design	$270,000	$240,000
Prototype Hardware	225,000	250,000
Tooling	100,000	40,000
Ground Tests	50,000	70,000

Test Facility (100-ft height) $300,000

SCHEDULE


Months
0
6
12
18
24
30
Single-Point Hoist
Design
Fabrication
Development Tests
Endurance
Environmental
Static
Four-Point Hoist
Design
Fabrication
Development Tests
Endurance
Environmental
Static
System Test
Hook & Swivel Assemblies
Design
Fabrication
Qualification Test

CONCLUSIONS

1. A separate function system that incorporates a mechanically driven single-point hoist and four hydraulically driven multi-point hoists best meets the external cargo handling requirements of a 40,000-pound-payload, single-rotor heavy lift helicopter on the basis of reliability, safety, maintainability, redundancy, versatility, and technical confidence.

2. The single- plus four-point system described above weighs 4974 pounds. For missions requiring single-point load suspension, the four-point hoists are readily removable, making the cargo handling system weight chargeable to aircraft weight empty 2738 pounds. Similarly, for four-point missions removal of single-point components results in a cargo system weighing 2704 pounds.

3. The mechanically driven single-point plus the hydraulically powered four-point hoist system has a capability of growth to 50,000 pounds. All major components of both hoists except the cable drum have adequate margins of safety to accommodate a 25-pct growth. The single-point system can be up-rated to 50,000 pounds for 170 additional pounds. The weight increase for the four-point system is 100 pounds.

4. With the use of a hydroelectrical feedback control system, the total cable length error (difference between individual cable lengths) in 50 feet of travel of the four-point system is 7-3/4 inches. This error may be further reduced to 4 inches by a one-time calibration procedure. A study should be undertaken to determine if these limits are within acceptable limits for equipment in the U.S. Army inventory.

5. The hydroelectrical feedback system proposed for the four-point system lends itself to evaluation and development in model form. Further effort in this area is recommended.

6. This investigation has led the writers to believe that acquisition and release of loads carried by four-point suspension will, for the most part, be accomplished with aircraft on the ground or from a "wheels light" hover. It is therefore concluded that an analysis of mission requirements is in order to determine if some other four-point cable length might better meet the heavy lift multi-point requirement. The limited investigation permitted within the scope of this contract has led the investigators to believe that a major portion of heavy lift helicopters should be equipped with powered four-point hoists with a maximum capability not exceeding 20 feet. Further study in this area is definitely required.

7. The use of manually actuated control valves which divert fluid from the aircraft utility hydraulic system to provide the power to operate the cable free reel clutches in the single- and the four-point hoists meets the requirement for a pilot actuated manual load release system.

8. In-flight release of loads under emergency conditions is more safely accomplished in the single-point mode (for an equal number of release components) where electrical, free reel, auto touchdown, and two explosive release systems are provided. For the four-point system, only the use of the tandem-dual cable cutters is recommended for in-flight release of loads under emergency conditions. The use of the free reel system to release loads in the four-point mode would be practical if the primary release system failed at the time that the aircraft was hovering just off the ground at the drop site, however.

9. The evaluation of current production cargo hook-swivel assemblies design indicates that further efforts are required to achieve the degree of reliability required. The swivel design and method of sealing the slip ring as proposed herein should add appreciably to the reliability of these components. A comprehenisve program to study and evaluate the reliability of pilot controlled load release would be invaluable to the aerial cargo handling system designer.

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APPENDIX I
SURVEY OF MILITARY VEHICLES

The 94 vehicles listed in Table XXXVIII represent the U.S. Army equipment considered in the design of the cargo handling system of this study. It is based on a survey conducted by the U.S. Army Combat Developments Command (Reference 14).

An item number has been assigned to each piece of equipment to provide a more convenient form of reference within the report. This procedure was necessary because the official "line number" of all the equipment was not known.

The equipment is listed in order of increasing weight. To prevent the table from becoming unwieldy, a special set of appreviations was devised. These abbreviations, listed below, apply only to Table XXXVIII.

List of Abbreviations

ABN.	airborne
AMB.	ambulance
AMMO.	ammunition
AMPH.	amphibious
ARMD.	armored
ASLT.	assault
AUX.	auxiliary
AVLB.	assault vehicle launch bridge
CARR.	carrier
C-C	combat weight
CGO.	cargo
CMD.	command
DRVN.	driven
DSL.	diesel
ENGR.	engineers
EQUIP.	equipment
EXP.	experimental
FT.	full track
GAL.	gallon
GEN.	generator
H	height, inches
HOW.	howitzer
IND.	industrial

L	length, inches
LT.	light
LWB.	long wheel base (trailer)
mm	millimeter
MED.	medium
MTD.	mounted
MTZD.	motorized
MULT.	multiple
OBS.	observation
PERS.	personnel
PROP.	propelled
RD.	road
RECONN.	reconnaissance
R/H	reduced height
RKT.	rocket
SP.	self-propelled
SPD.	speed
STLR.	semi-trailer
SUP.	supply
SVC.	servicing
SWB.	short wheel base
T	ton
TLR.	trailer
TRACT.	tractor
TRANSP.	transporter
TRK.	truck
TRKD.	tracked
VEH.	vehicle
W	width, inches
WHLD.	wheeled
WKR.	wrecker
WPH.	weapons
WWN.	with winch
XLWB.	extra long wheel base
YD.	yard

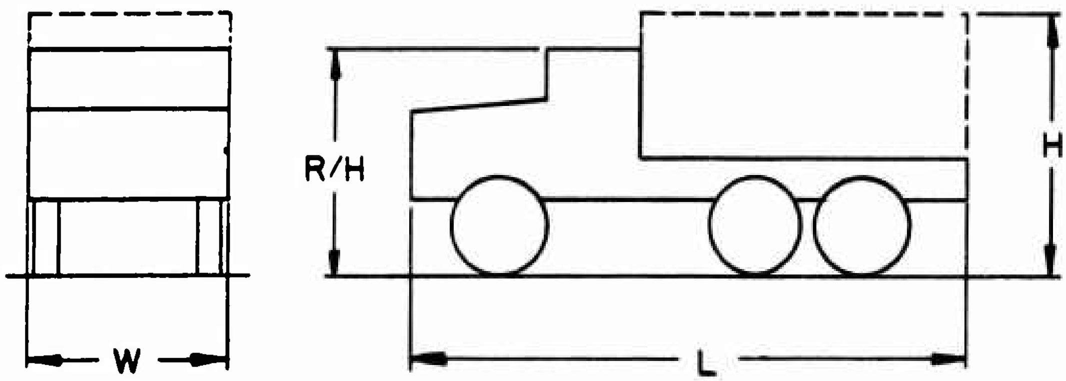


Figure 65. Typical Military Vehicle.

TABLE XXXVIII
LIST OF MILITARY VEHICLES

Item No.	Model No.	Designation	Weight Net (Tons)	Weight C-C (Tons)	Dimensions L (Inches)	Dimensions W (Inches)	Dimensions H (Inches)	Dimensions R/H (Inches)
1	M-100	TLR. AMPH. CGO. 1/4-T	0.283	0.533	109	57	42	-
2	M-14	CART RKT.TRANSPT. 318 mm	0.340	0.340	221	58	31	58
3	M-274	CARR.LT.WPH. 1/2-T	0.398	0.985	117	46	28	-
4	-	PROP.,EQUIP. AUX. HOW.	0.449	-	-	-	-	-
5	M-91	LAUNCHER, MULT. RKT. 115 mm	0.625	0.625	152	90	67	-
6	XM-34	LAUNCHER, RKT. 318 mm	0.659	1.117	126	62	50	13
7	M-101	TLR.,CGO. 3/4-T	0.670	1.420	147	74	85	53
8	OH-13	HELICOPTER, OBS.	0.859	1.400	365	90	143	112

TABLE XXXVIII (continued)

Item No.	Model No.	Designation	Weight Net (Tons)	Weight C-C (Tons)	Dimensions L (Inches)	W	H	R/H
9	M-151	TRK. UTILITY	1.137	1.417	132	63	71	53
10	M-332	TLR., AMMO.	1.175	2.225	153	86	56	-
11	T-3	TRANSPORTER, LIQUID 1000 GAL.	1.200	4.200	155	102	64	-
12	M-200A1	CHASSIS TLR. GEN. 2-1/2-T	1.205	4.705	165	93	40	-
13	M-149	TLR., WATER 1-1/2-T	1.300	3.948	157	80	73	-
14	M-105A2	TLR., CGO. 1-1/2-T	1.325	2.700	166	83	98	58
15	M-38AIC	TRK., UTILITY 1/4-T	1.333	1.733	109	57	42	-
16	-	TLR., BASIC UTILITY 2-1/2-T	1.350	4.850	198	98	44	-
17	M-170	TRK., AMB. 1/4-T	1.482	1.767	155	61	79	-
18	XM-102	HOW., 105 mm TOWED- SP.	1.530	1.530	-	-	-	-
19	M-242	HOW., 105 mm TOWED	2.490	6.350	236	85	62	-
20	UH-1B	HELICOPTER, UTILITY	2.328	3.600	481	103	157	90
21	UH-1D	HELICOPTER, UTILITY	2.328	3.600	481	103	157	90
22	M-37	TRK., CGO. 3/4-T	2.850	3.600	185	74	90	64
23	M-37	TRK., CGO. 3/4-T WWN.	3.000	3.725	190	74	90	64
24	M-146	STLR., VAN SHOP	3.400	6.000	276	95	136	-
25	-	TRK., FORK LIFT 2-T	3.400	3.400	-	-	-	-
26	M-118A1	STLR., STAKE 6-T	3.570	9.570	276	95	104	38

TABLE XXXVIII (continued)

Item No.	Model No.	Designation	Weight Net (Tons)	Weight C-C (Tons)	Dimensions L (Inches)	Dimensions W (Inches)	Dimensions H (Inches)	Dimensions R/H (Inches)
27	M-43	TRK., AMB. 3/4-T	3.585	4.275	206	75	92	-
28	M-119A3	STLR., VAN, CGO. 6-T	3.590	9.590	275	96	135	-
29	M-405	HANDLING UNIT, 762 mm RKT.	4.315	4.315	338	96	130	81
30	-	TLR., LOW BED 8-T	4.915	12.915	281	102	58	-
31	M-9	BULLDOZER	5.000	5.000	146	36	78	-
32	OV-1	AIRPLANE, OBS.	5.328	6.659	525	110	156	-
33	M-273	TRK., TRACT. 2-1/2-T SWB.	5.590	5.590	228	94	98	81
34	M-345	TLR., FLAT BED 10-T	5.630	15.630	536	98	56	-
35	M-48	TRK., TRACT. 2-1/2-T LWB.WWN.	5.921	5.921	254	94	98	82
36	M-35 (XM-410E)	TRK., CGO, 2-1/2 -T LWB.	6.233	11.408	262	96	112	86
37	M-56	GUN PT., 90 mm ABN.	6.250	8.750	241	88	88	-
38	M-114A1	HOW., 155 mm TOWED	6.350	6.350	288	96	81	-
39	M-35	TRK., CGO 2-1/2-T DWB-WWN.	6.440	11.790	276	96	112	86
40	M-129	STLR., VAN SUP. 12-T	6.750	18.750	345	96	140	-
41	M-49C	TRK., TANK, FUEL SVC. 2-1/2-T	6.978	6.978	262	96	98	90
42	-	SCRAPER, TOWED 7-1/2 YD.	7.060	7.060	-	-	-	-
43	M-49C	TRK., TANK, FUEL SVC. 2-1/2-T WWN.	7.200	7.200	277	99	130	-

TABLE XXXVIII (continued)

Item No.	Model No.	Designation	Weight Net (Tons)	Weight C-C (Tons)	Dimensions L (Inches)	Dimensions W (Inches)	Dimensions H (Inches)	R/H
44	-	LOADER, SCOOP TYPE 1-1/2 YD^3	7.200	7.200	209	84	81	-
45	M-127A1	STLR., STAKE 12-T	7.200	19.400	345	97	109	58
46	M-131A3C	STLR., TANK, FUEL SVC.	7.400	7.400	353	98	110	-
47	M-172A1	STLR. LOW BED 25-T	7.430	32.430	141	115	68	-
48	MT 2D	GRADER, RD. MTZD. DSL. DRVN.	7.460	7.460	264	81	85	-
49	M-114A1	ARMD. RECONN. CARRIER -FT.	7.500	7.500	169	92	80	-
50	M-313	STLR. VAN, EXP. 6-T	7.500		323	98	134	-
51	M-220	TRK., VAN SHOP 2-1/2-T	7.543	10.043	267	96	131	-
52	M-342	TRK., DUMP 2-1/2 -T	7.583	10.083	273	96	101	83
53	M-342	TRK., DUMP 2-1/2 -T WWN.	7.790	10.290	273	96	100	100
54	M-109	TRK., VAN, SHOP 2-1/2-T WWN.	7.823	10.291	277	99	130	-
55	-	TRACTOR, FT., LOW SPEED, DSL. DRVN., LT.	7.988	7.988	175	99	78	-
56	CH-47A	HELICOPTER, CGO., MED.	8.000	16.500	600	145	222	-
57	M-129	STLR., VAN SUP. 12-T	8.010	30.010	345	96	140	-
58	-	TRACTOR, WHLD. IND. DSL. DRVN. LT.	8.050	8.050	194	96	90	-
59	-	TRK., FORK LIFT 3-T	8.400	8.400	-	-	-	-

TABLE XXXVIII (continued)

Item No.	Model No.	Designation	Weight Net (Tons)	Weight C-C (Tons)	Dimensions L (Inches)	W	H	R/H
60	M-270A1	STLR., LOW BED WKR. 12-T	8.750	20.750	597	97	121	80
61	M-52	TRK., TRACTOR 5-T SWB.	9.200	9.200	258	97	107	87
62	M-292	TRK., VAN EXP. 2-1/2-T	9.500	12.000	329	97	139	-
63	M-54 (XM-656)	TRK., CGO. 5-T LWB.	9.616	14.616	299	97	116	86
64	M-113	CARRIER, PERS. FT.	9.878	11.308	192	106	80	-
65	M-108	TRK., WKR., CRANE 2-1/2-T WWN.	9.893	1C.143	303	96	100	-
66	M-54 (XM-656)	TRK., CGO., 5-T LWB, WWN.	9.973	14.973	314	97	116	86
67	M-577	CARRIER, CMD.POST LT. TRKD.	10.700	11.650	192	106	106	-
68	M-51	TRK., DUMP 5-T WWN.	11.333	16.333	282	98	111	88
69	XM-106	MORTAR, SP.,FT. 4.2 in.	12.538		192	106	80	-
70	M-60	TRK., WKR.,LT. 2-1/2-T	11.980	12.980	303	96	101	-
71	M-78	HEAT & TIEDOWN UNIT 762 mm RKT.	12.032	12.032	370	96	95	-
72	-	LOADER, SCOOP TYPE DSL. DRVN.1-1/2 YD3	12.200	12.200	248	105	97	-
73	-	GRADER, RD.,MTZD., DSL. DRVN.	12.390	12.390	311	96	111	93
74	M-139	TRK., STAKE 5-T BRIDGE TRANSP.	13.400	13.400	369	114	114	-
75	M-123C	TRK., TRACTOR 10-T	14.200	14.200	280	114	113	92

TABLE XXXVIII (continued)

Item No.	Model No.	Designation	Weight Net (Tons)	Weight C-C (Tons)	Dimensions L (Inches)	W	H	R/H
76	CL-60	AVLB.	14.300	14.300	338	158	73	-
77	-	LOADER, SCOOP TYPE DSL. DRVN., 2-1/4 YD^3	14.414	14.414	248	105	97	-
78	M-551	ARMD. RECONN./ ABN. ASLT. VEHICLE	15.000	15.000	252	115	95	-
79	M-115	HOW. 8-in. TOWED	15.288	15.288	432	112	108	-
80	-	TRACTOR, FT. DSL. DRVN. LOW SPD., MED.	15.666	15.666	196	116	88	-
81	M-162A1	STLR. LOW BED 60-T	16.348	76.348	441	144	81	-
82	M-246	TRK., TRACT. XLWB. WWN.5-T	16.415	16.415	352	98	132	89
83	M-62	TRK., WKR., MED., 5-T WWN.	16.700	16.700	310	97	103	-
84	M-386	LAUNCHER, 762 mm TRK., MTD.	17.291	17.291	389	114	105	-
85	M-572	HANDLING UNIT, 318 mm TRK. MTD.	19.850	19.850	339	96	97	-
86	M-84 (XM-106)	MORTAR, SP., FT., 4.2 in.	20.561	20.561	221	129	109	-
87	M-15A2	STLR. TRK., TRANSPORTER	21.300	71.300	462	146	105	-
88	(XM-551)	TANK COMBAT, FT. LT. GUN 76 mm (ARMD. RECONN./ABN. ASLT. VEH.)	25.400	25.900	280	126	122	108
89	M-44 (M-109)	HOW. SP., FT., 155 mm	29.000	29.000	325	140	134	127

TABLE XXXVIII (continued)

Item No.	Model No.	Designation	Weight Net (Tons)	Weight C-C (Tons)	Dimensions L (Inches)	W	H	R/H
90	M-55 (M-110)	HOW.SP. FT. 8 in.	45.000	45.000	325	140	146	117
91	M-60	TANK, COMBAT FT. 105 mm	47.150	47.150	366	143	126	-
92	M-60	LAUNCHER, AVLB	47.650	47.650	274	143	127	-
93	M-102	COMBAT, ENGR. VEH.	51.800	55.000	337	148	122	-
94	M-88	TANK RECOVERY VEH. MED.	54.000	56.000	326	135	127	-

APPENDIX II
MECHANICAL VARIABLE SPEED DRIVE

INTRODUCTION

A mechanical variable speed drive concept, designed and developed by the Lycoming Division, AVCO Corporation, was investigated as an alternate to the clutch-reverser unit of page 137 as the drive for the mechanically powered single-point hoists.

This concept provides an infinitely variable bidirectional output rotation mechanism. The unit proposed for the hoists of this study is a modified form of the traction mechanism employed in the constant speed drive units for A.C. generating systems used on the Navy A4E. Over 1000 of these units have been produced and approximately 500,000 operational hours have been accumulated.

DESCRIPTION

The proposed actuator combines an epicyclic gear differential coupled to an infinitely variable ratio traction transmission. This combination results in a unit capable of infinitely variable, stepless, bidirectional output rotation.

The variable ratio section consists of two flywheel members, called toroids, which are concentric with the drive shaft. Their dished surfaces form a toroidal space and contain four rolls, mounted in yokes and fastened to a fixed cage. A precalibrated load bolt squeezes the toroids against the rolls and provides the traction for power transmission. Speed ratios are changed by varying the angular position of the rolls with respect to the drive axis. The control rod motion can be linearized with respect to output speed if desired. The differential section is an epicyclic gear train and is so designed as to permit equal and opposite output speed. The planet cage rotates as a function of the position of the rolls in the ratio change section.

Table XXXIX gives the applicable design data of the unit. Figure 67, page 249, gives the output power and torque versus rpm relationship.

It should be noted that added cooling is required for this unit. It will be provided by an electric motor driven blower/heat exchanger unit of the type used on the CH-3C and CH-53A. This unit will be interlocked so as to function only when the variable speed drive unit is in operation. Figure 66 is included to show the physical dimensions of the complete unit, less heat exchanger.

It has been estimated that it would require approximately 16 months to complete the design, fabrication, and prototype developmental testing required prior to delivery.

TABLE XXXIX

DESIGN DATA,
MECHANICAL VARIABLE SPEED DRIVE

Input Speed	7000 rpm
Output Speed	Plus or minus 1000 rpm
Power Capability	225 HP maximum (lifting load) 115 HP maximum (lowering load)
Control Power (max)	.01 HP at maximum acceleration rate of 1300 rpm/sec
Control Force	45 lb at control rod for maximum acceleration
Efficiency	93 pct
Lubrication	MIL-S-81087 (Weps) Type 1
Oil Flow	20 lb/min
Cooling Requirements	500 Btu/min
Operating Temperature	-65°F to 350°F
Weight (Estimated)	
Dry weight with integral oil reservoir and pump	107 lb
Lubricant	6 lb

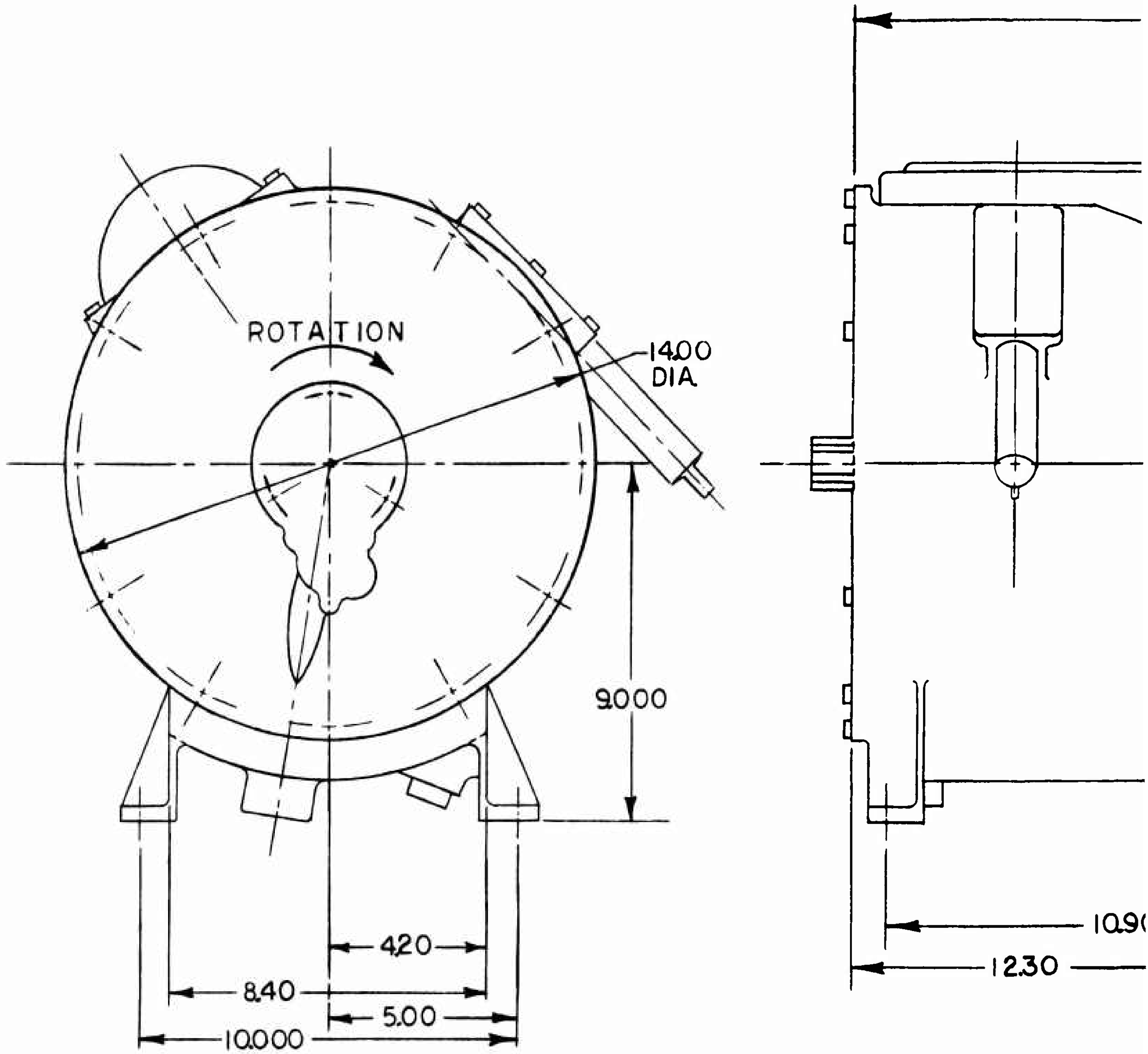
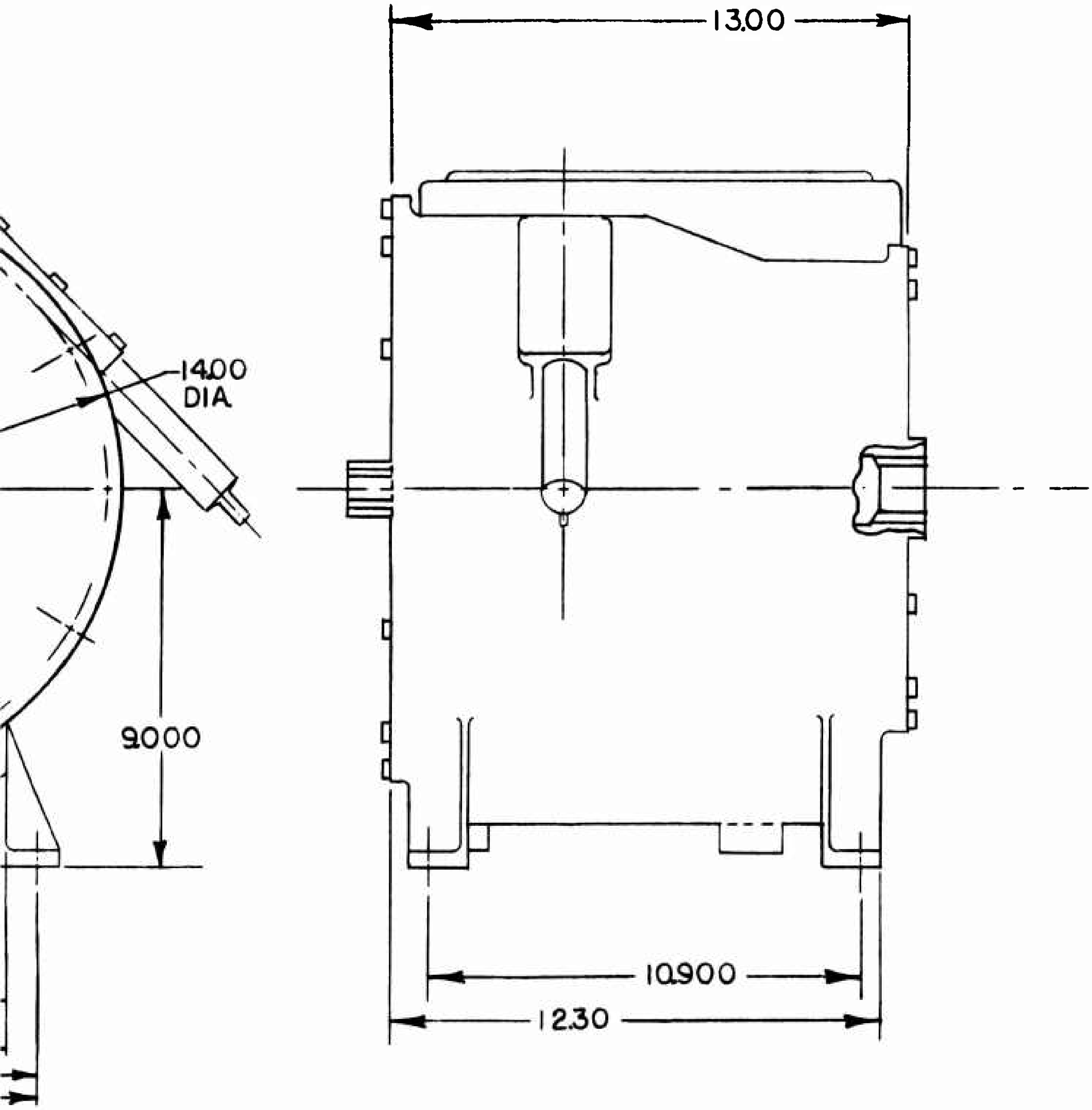


Figure 66. Variable Speed Drive.

A.



eed Drive.

B.

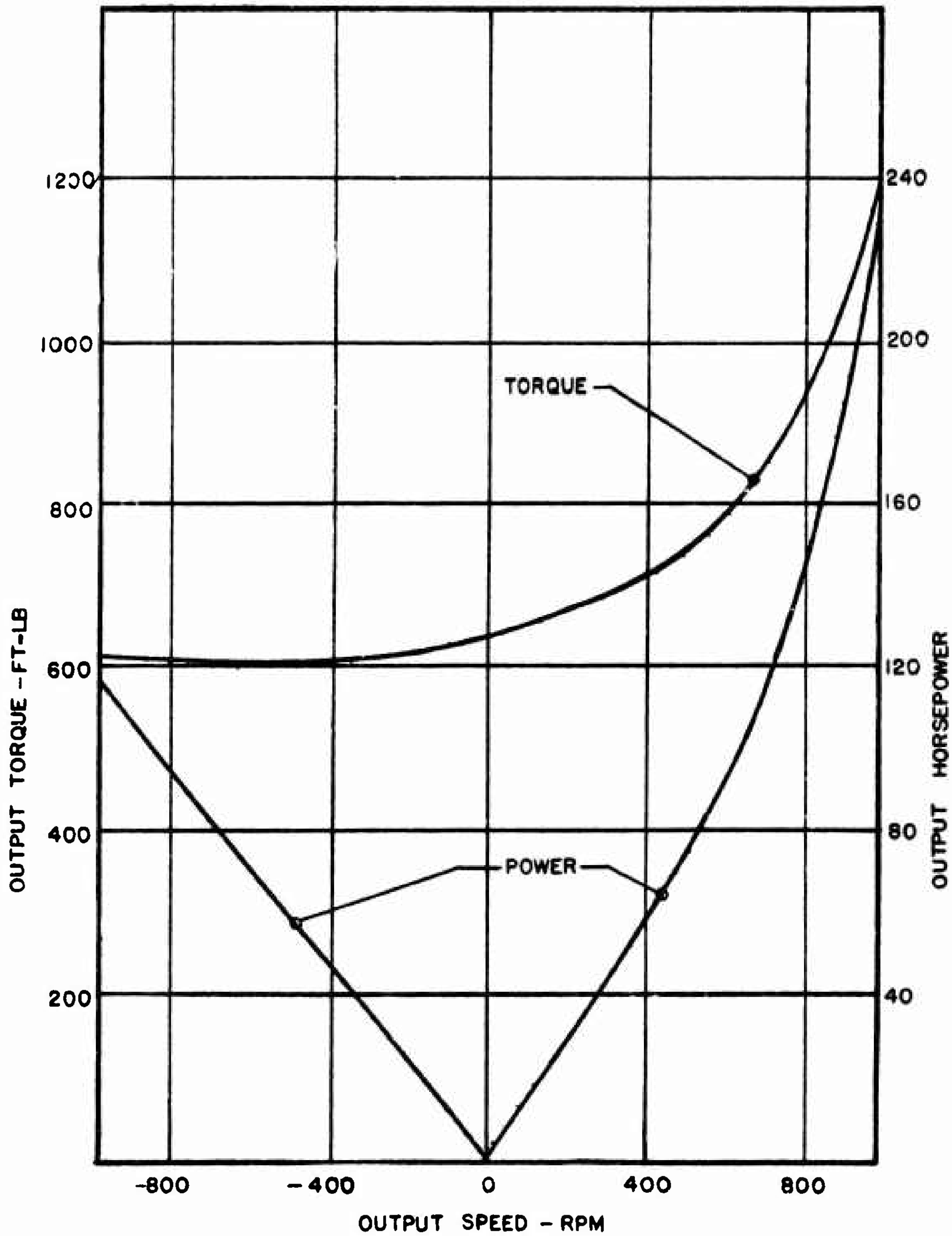


Figure 67. **Output Power and Torque vs Output Speed, Variable Speed Drive.**

APPENDIX III
CARGO HANDLING SYSTEM DRAWINGS

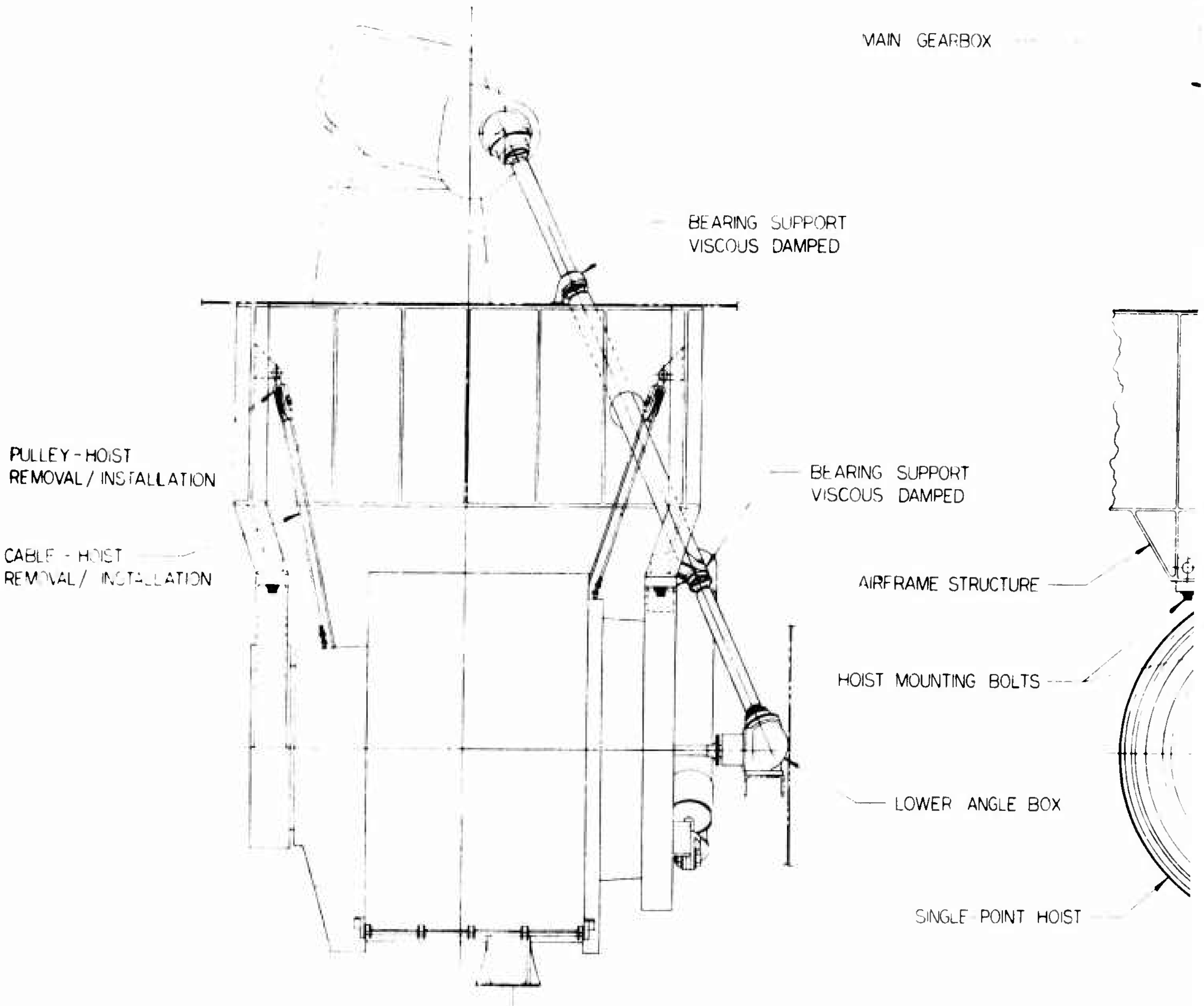
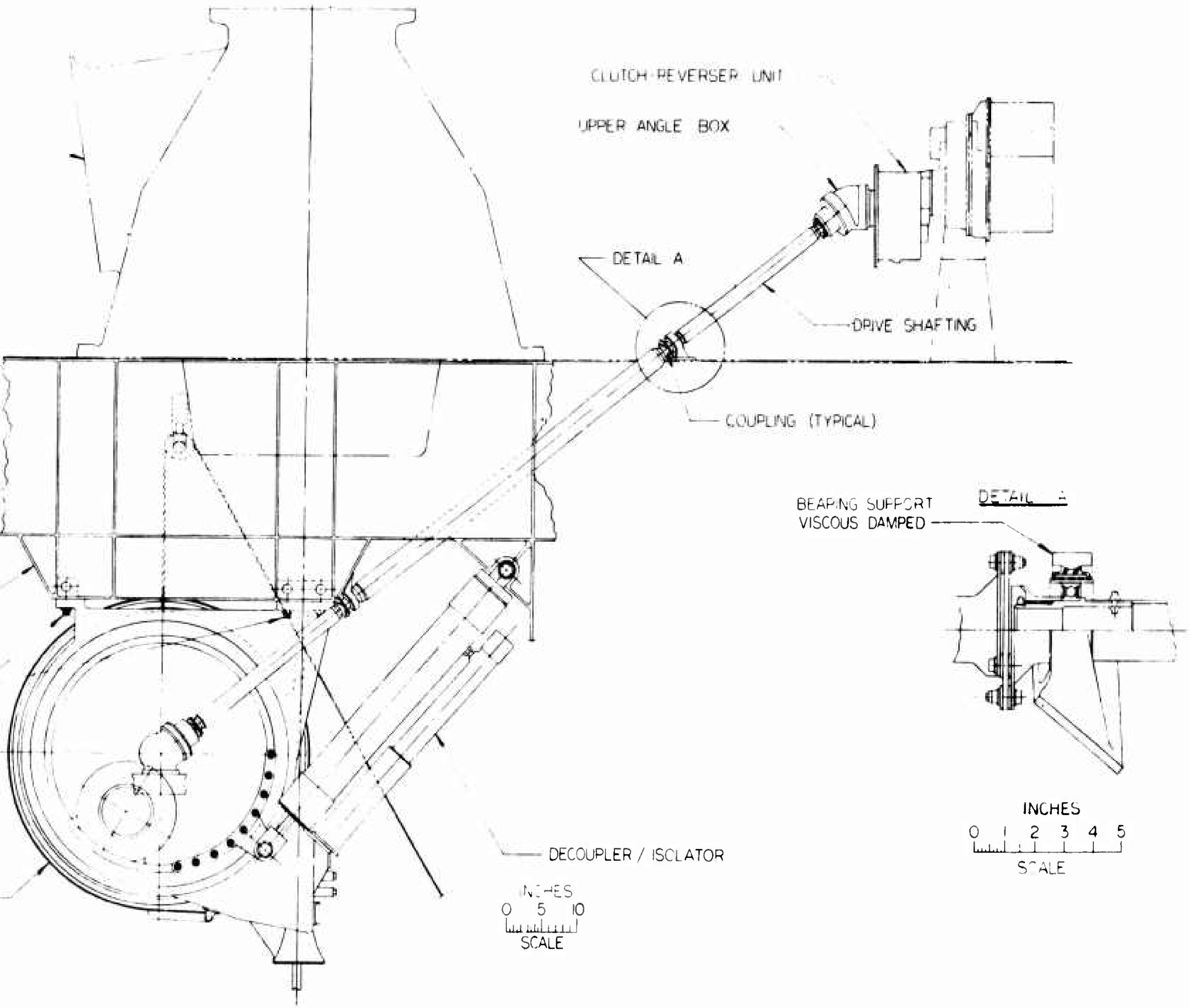
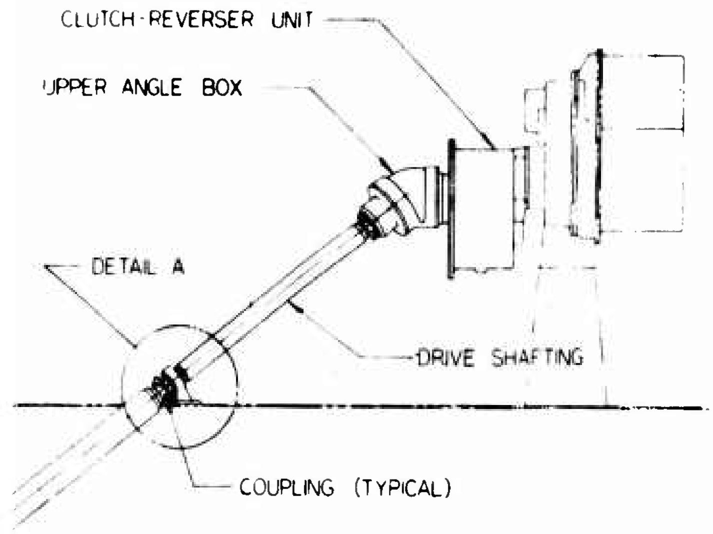


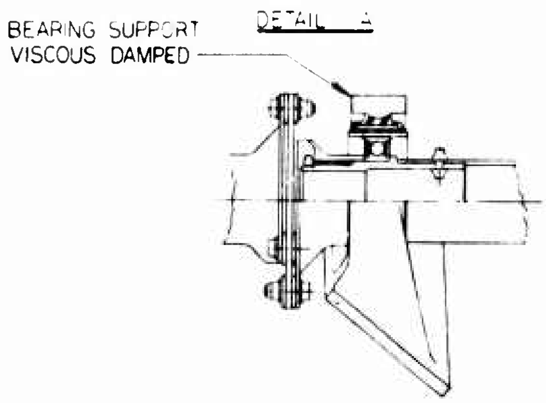
Figure 68. Single-Point Hoist Installation, Single-Rotor H.L.H.

A.


CLUTCH-REVERSER UNIT
UPPER ANGLE BOX
DETAIL A
DRIVE SHAFTING
COUPLING (TYPICAL)
DECOUPLER / ISOLATOR
INCHES
0 5 10
SCALE
BEARING SUPPORT
VISCOUS DAMPED
DETAIL A
INCHES
0 1 2 3 4 5
SCALE

B.

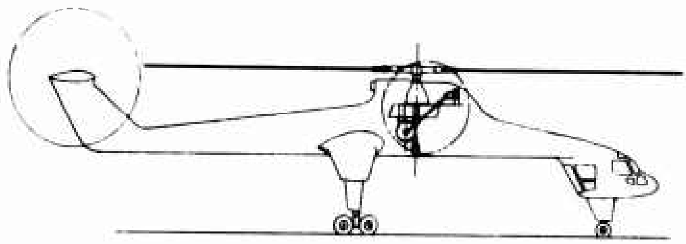




INCHES
0 1 2 3 4 5
SCALE

DECOUPLER / ISOLATOR

ES
10
E



INCHES
0 50 100
SCALE

e.

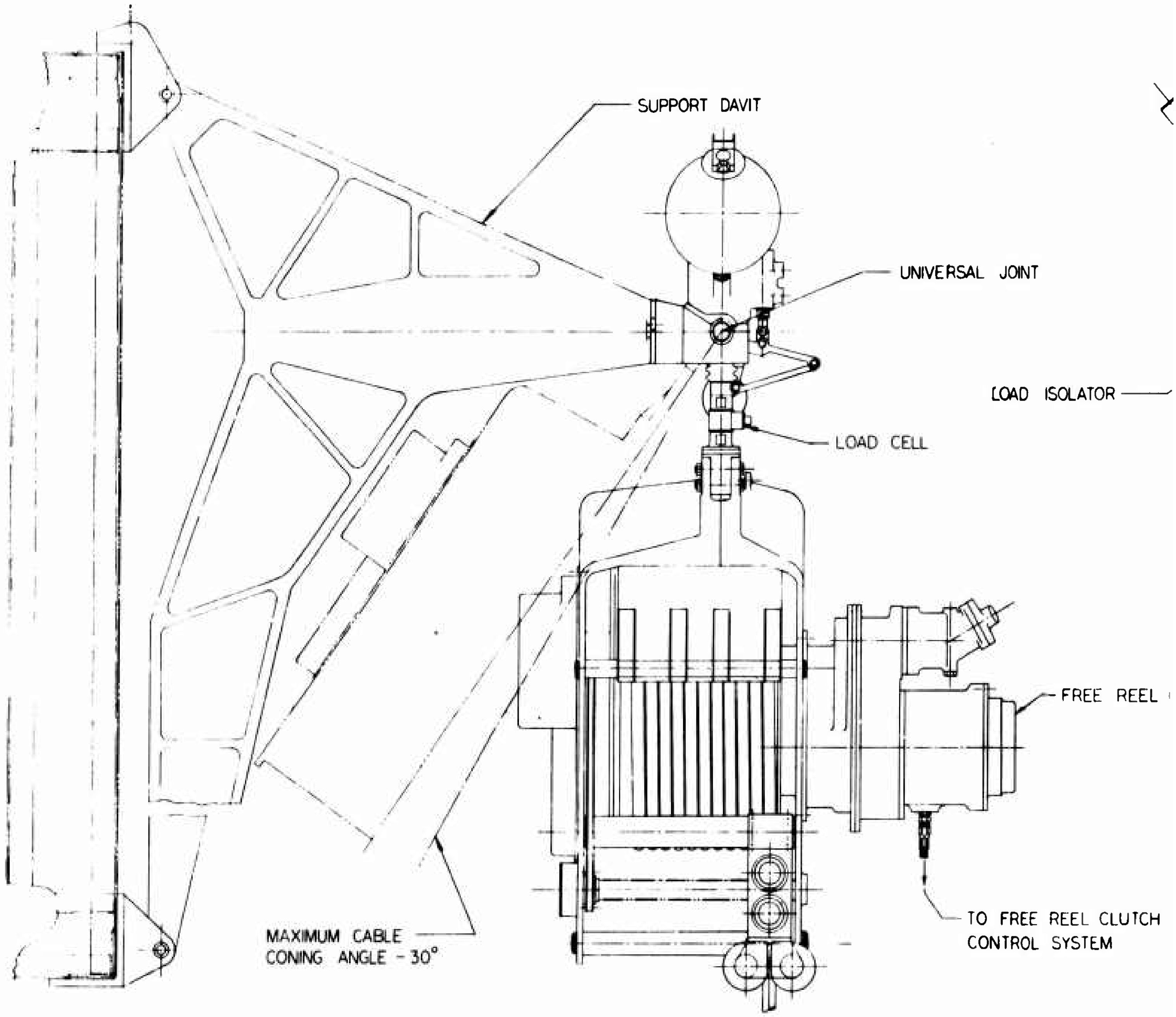
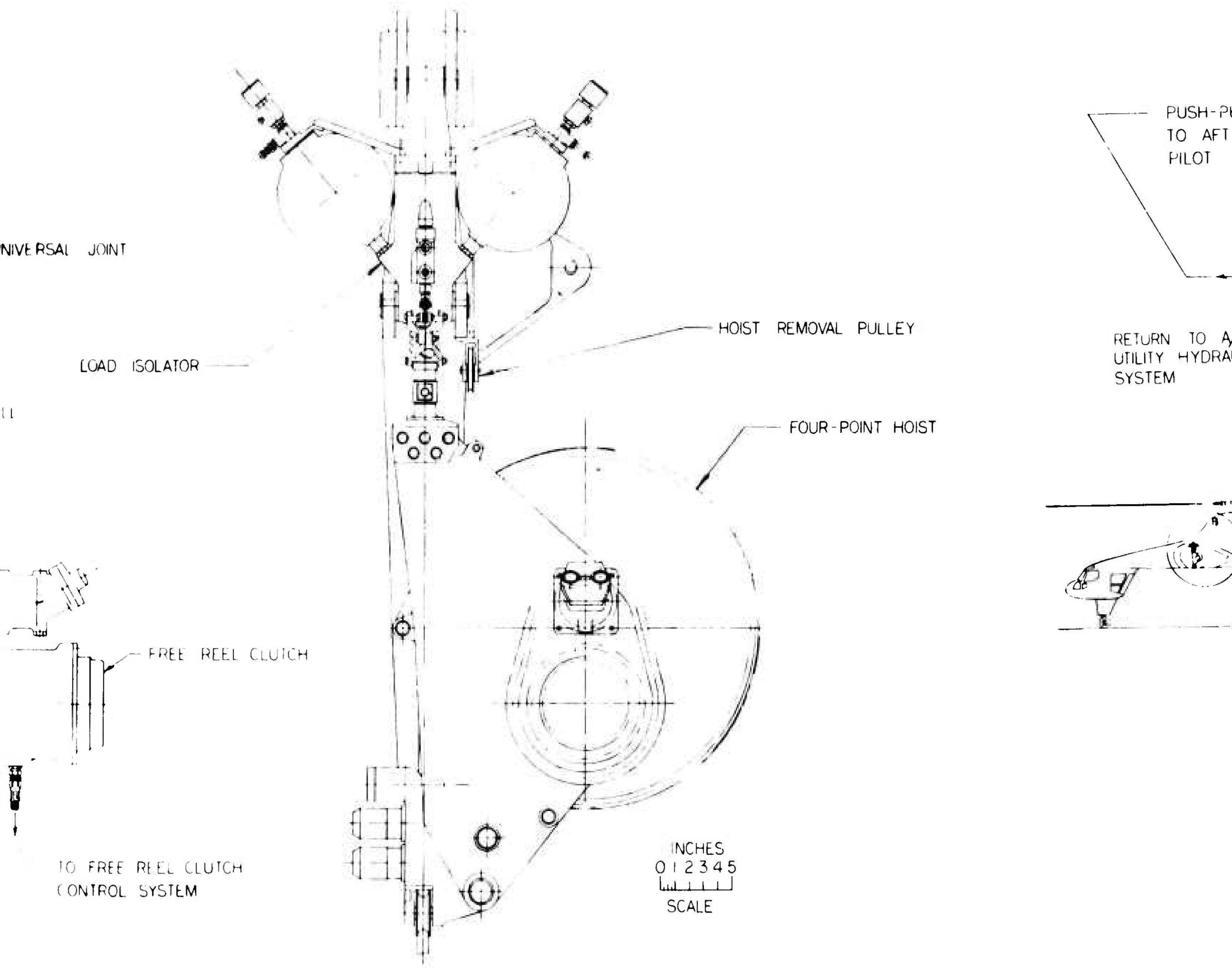


Figure 69. Four-Point Hoist Installation, Single-Rotor H.L.H.

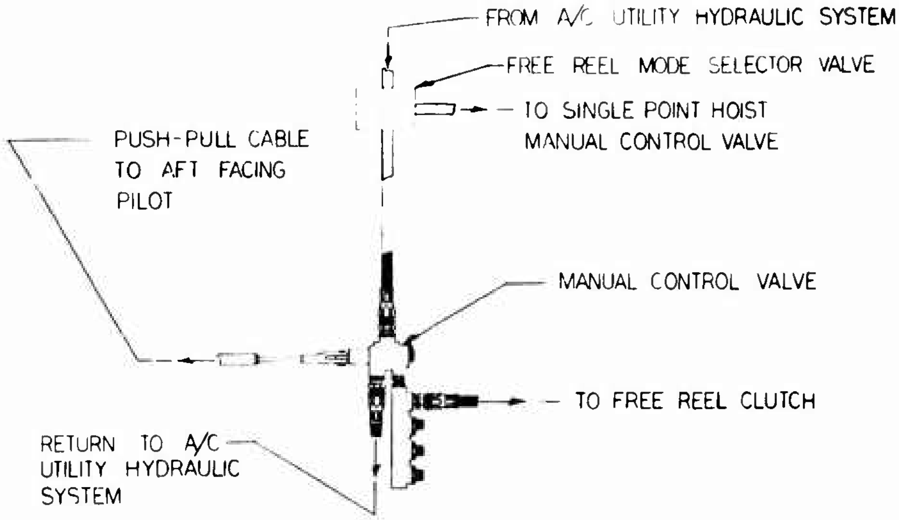
A.

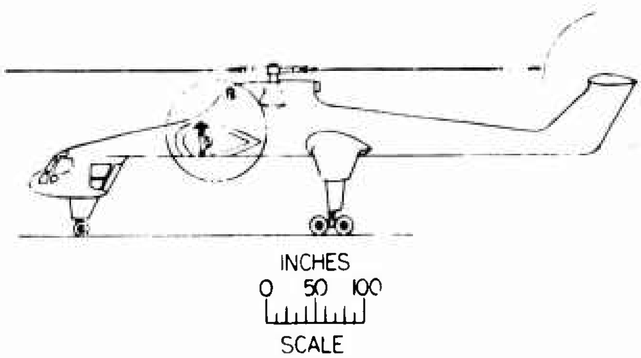

JOINT
LOAD ISOLATOR
HOIST REMOVAL PULLEY
FOUR-POINT HOIST
FREE REEL CLUTCH
TO FREE REEL CLUTCH
CONTROL SYSTEM
INCHES
0 1 2 3 4 5
SCALE
TO AFT
PILOT
RETURN TO
UTILITY
SYSTEM

B

EMOVAL PULLEY

FOUR-POINT HOIST





c.

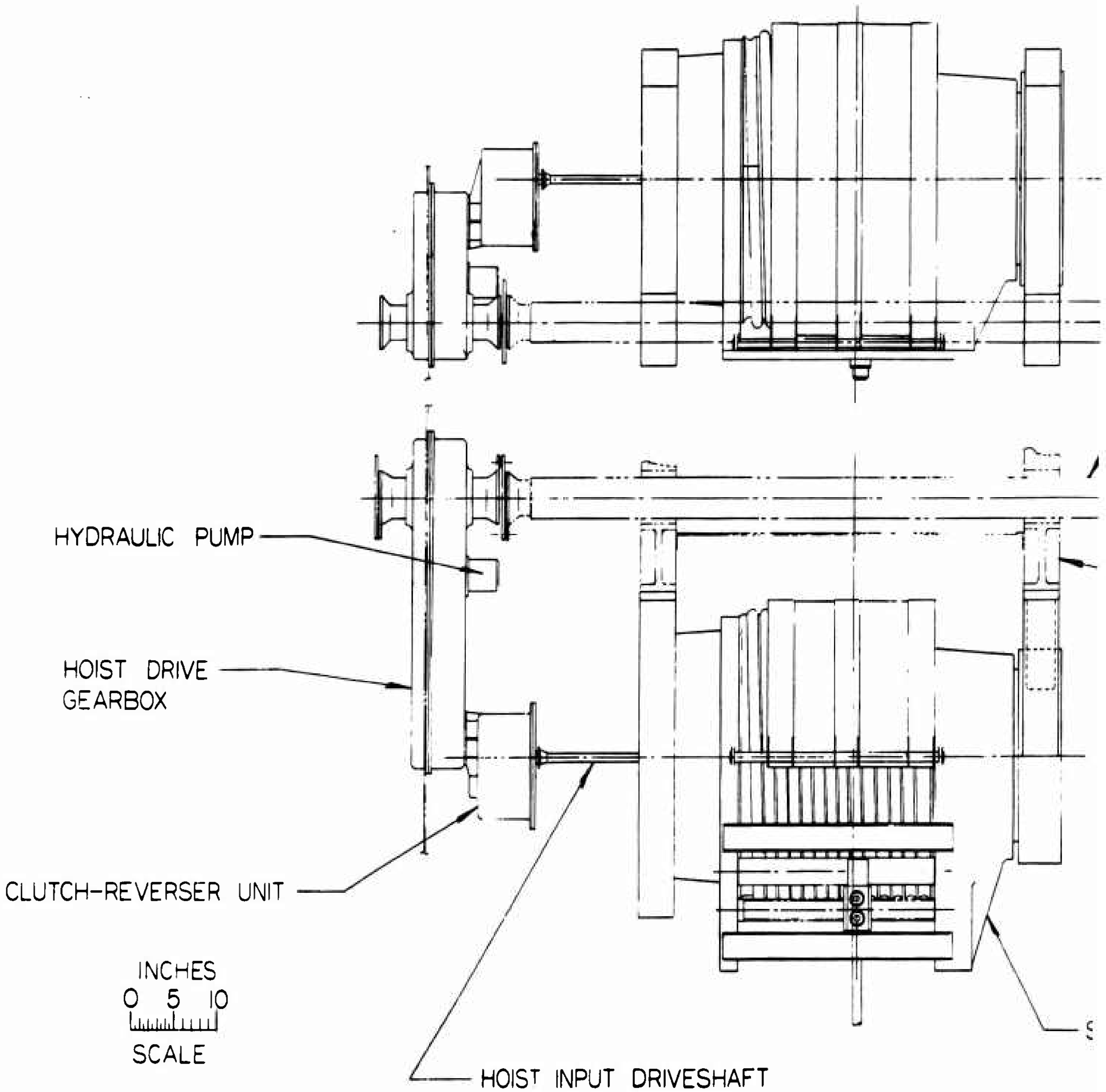
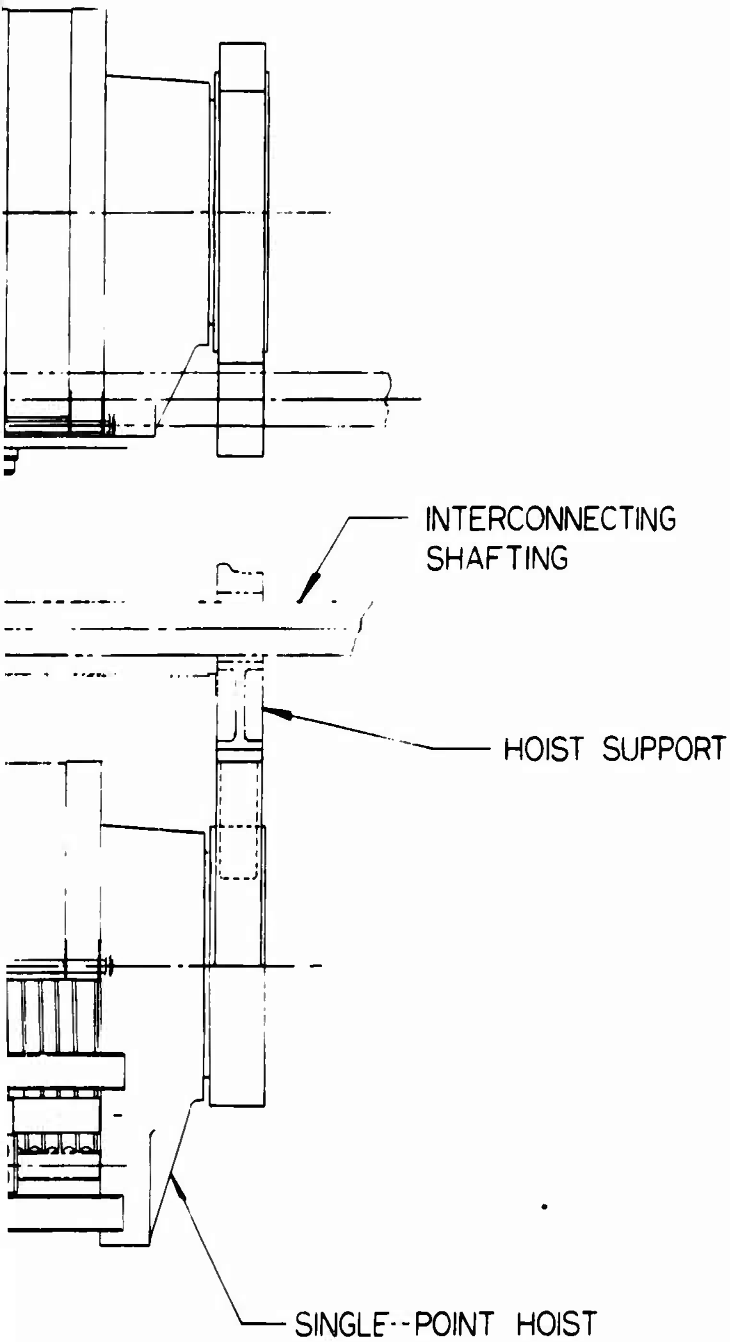
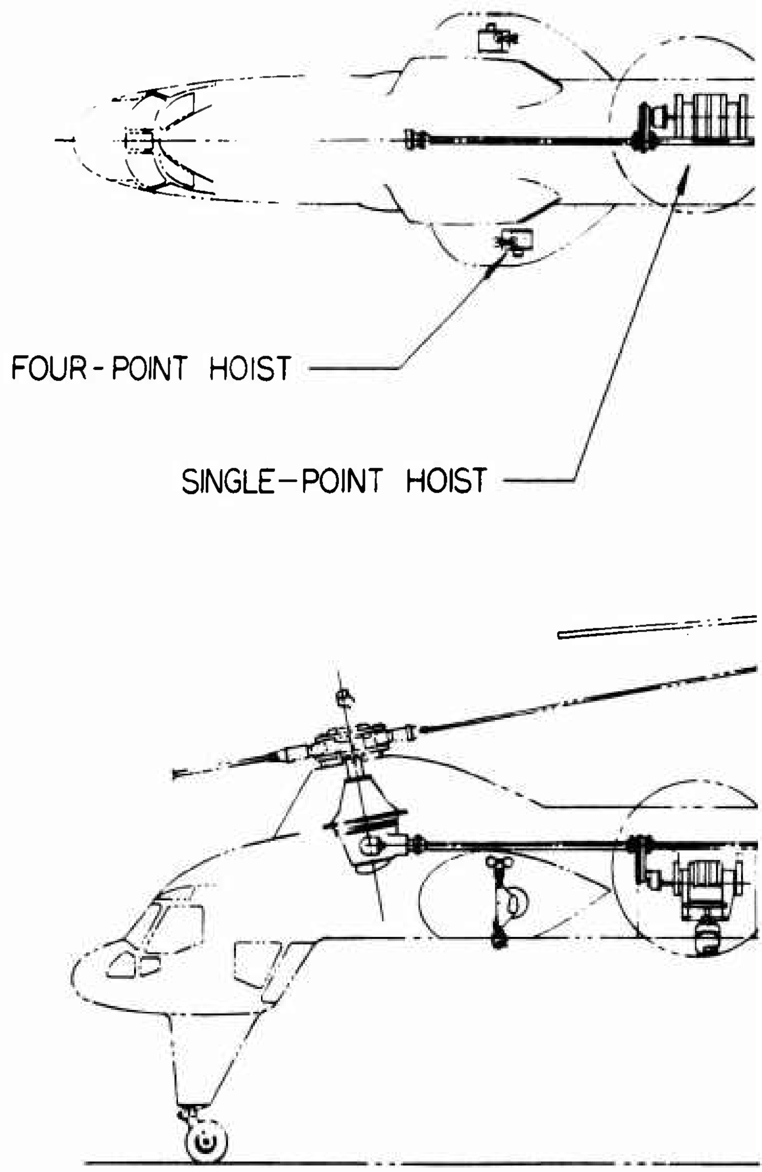


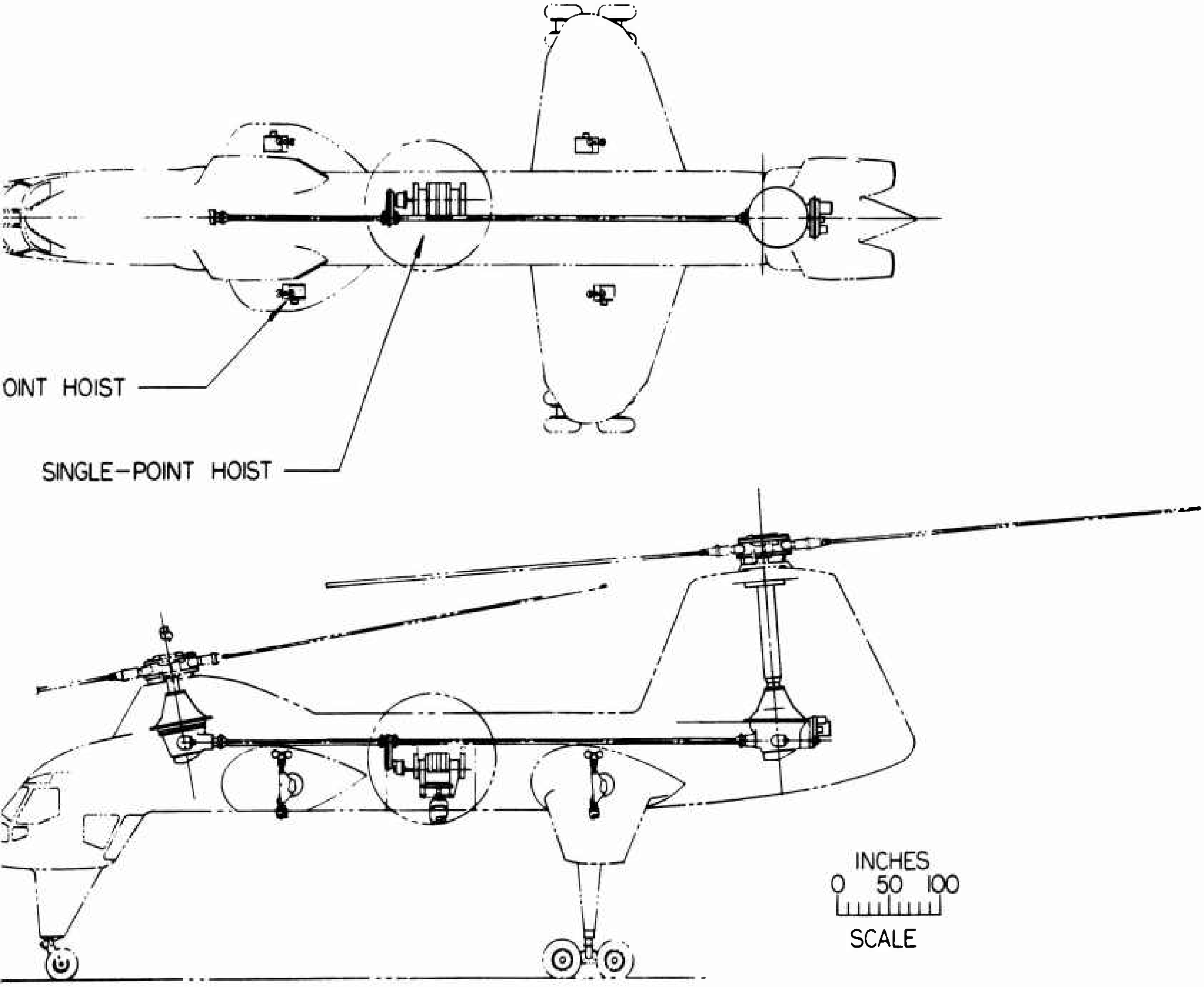
Figure 70. Single- Plus Four-Point Hoist Installation, Tandem-Rotor H.L.H.

A.


INTERCONNECTING SHAFTING
HOIST SUPPORT
SINGLE--POINT HOIST


FOUR-POINT HOIST
SINGLE-POINT HOIST

B.



C.

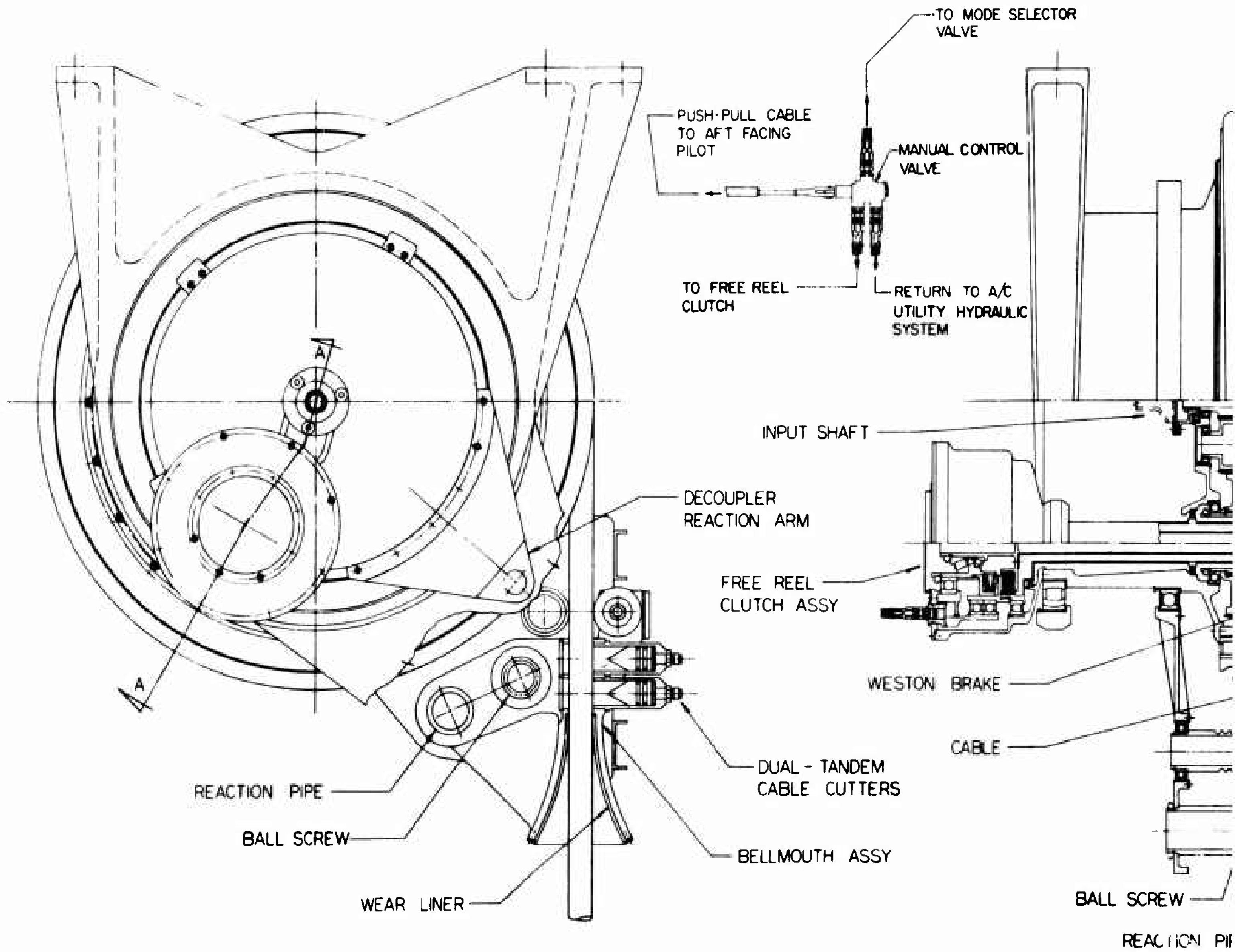
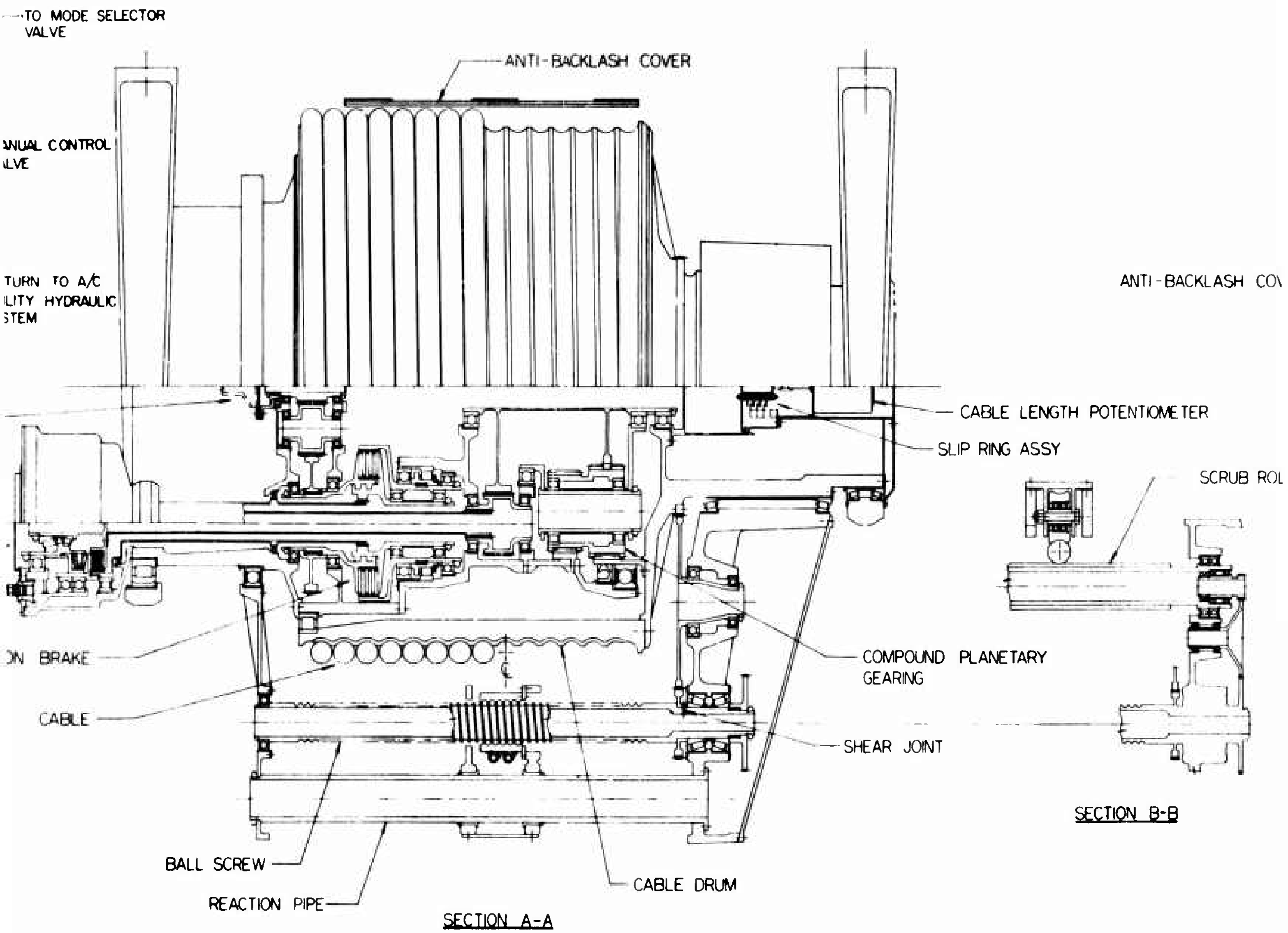
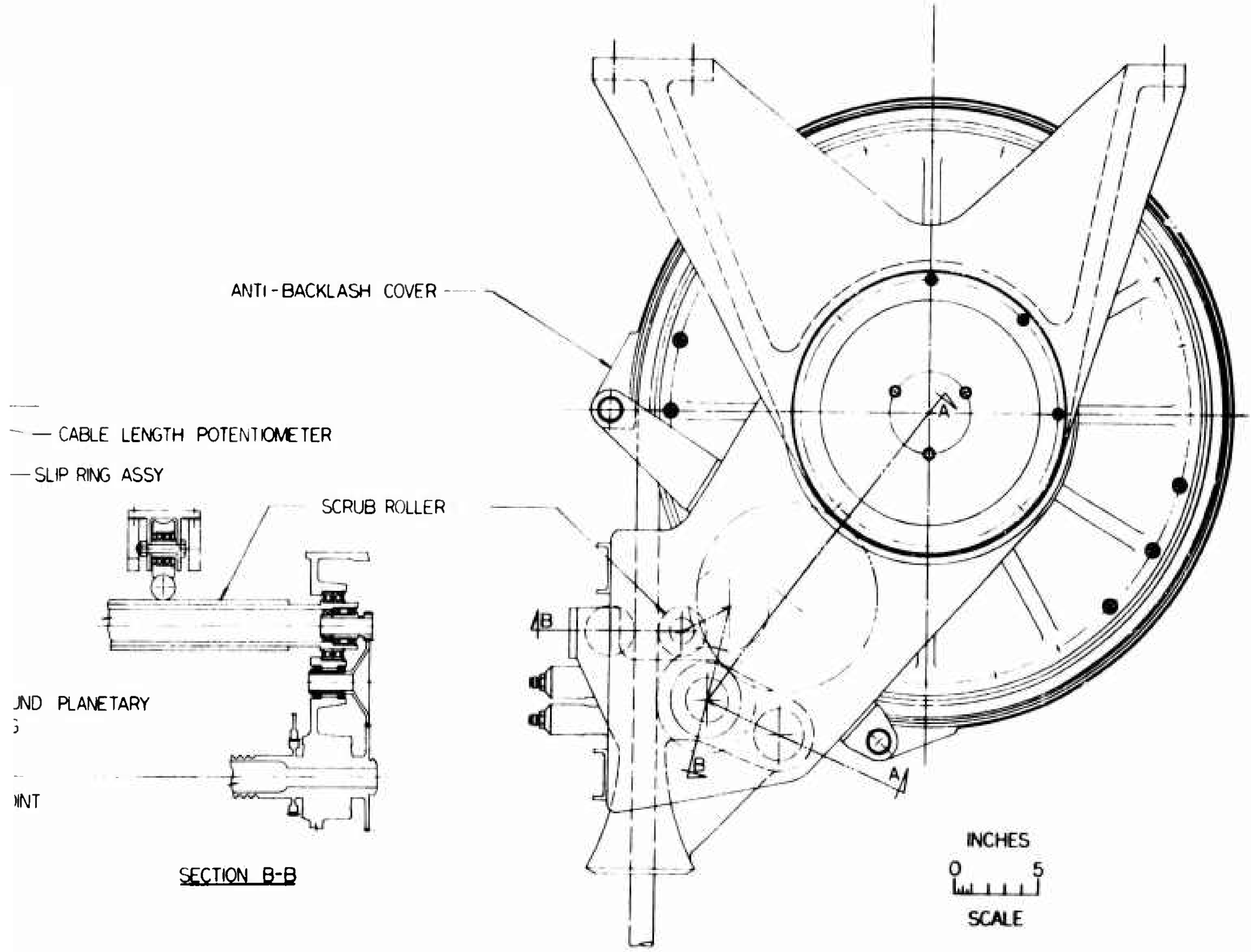


Figure 71. Single-Point Hoist.

A.


TO MODE SELECTOR VALVE
ANTI-BACKLASH COVER
ANUAL CONTROL
LVE
TURN TO A/C
LITY HYDRAULIC
STEM
ANTI-BACKLASH CO
CABLE LENGTH POTENTIOMETER
SLIP RING ASSY
SCRUB RO
ON BRAKE
CABLE
COMPOUND PLANETARY GEARING
SHEAR JOINT
BALL SCREW
REACTION PIPE
CABLE DRUM
SECTION A-A
SECTION B-B

B.


ANTI-BACKLASH COVER
CABLE LENGTH POTENTIOMETER
SLIP RING ASSY
SCRUB ROLLER
PLANETARY
SECTION B-B
A
B
INCHES
0
5
SCALE

C.

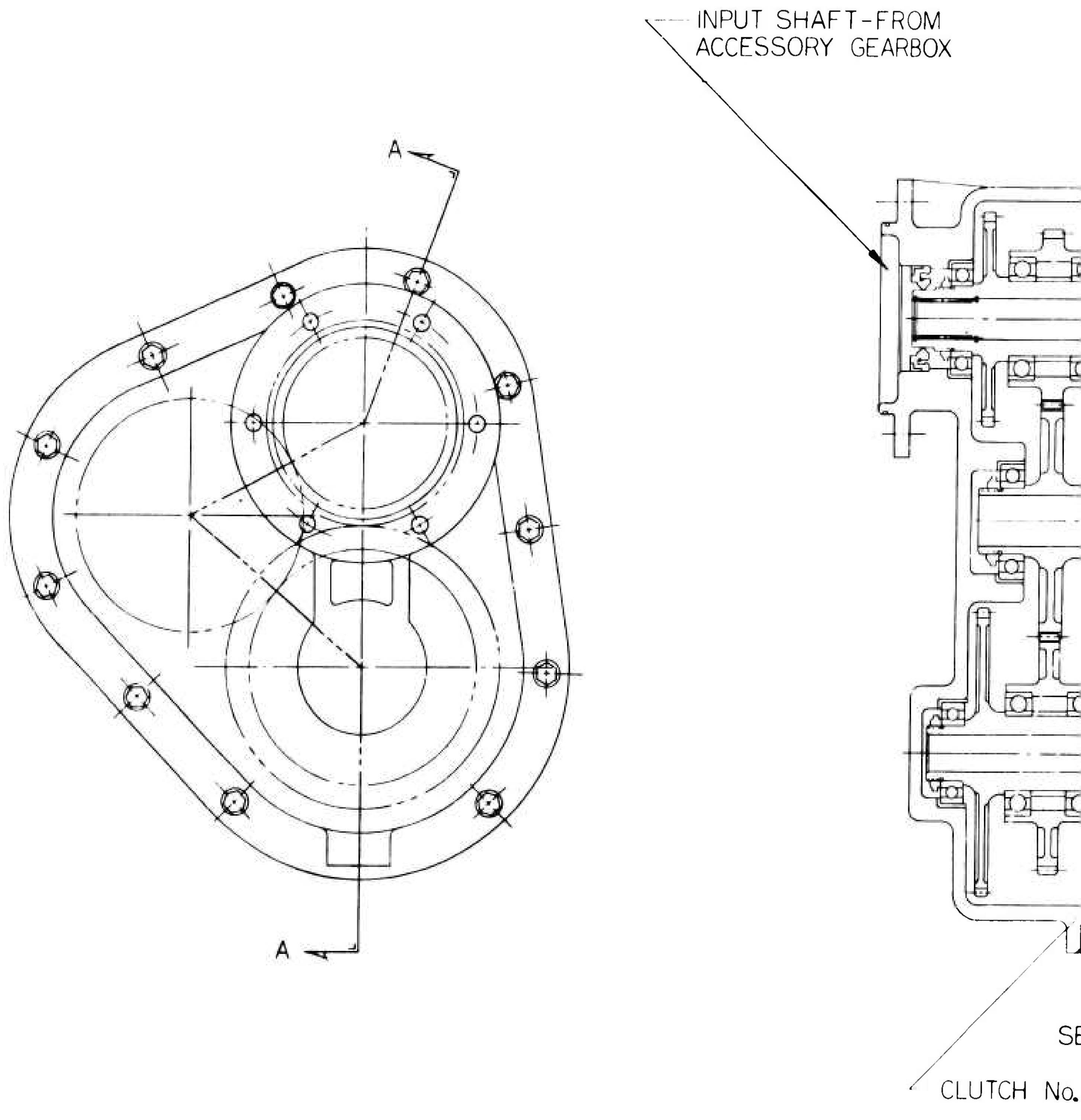
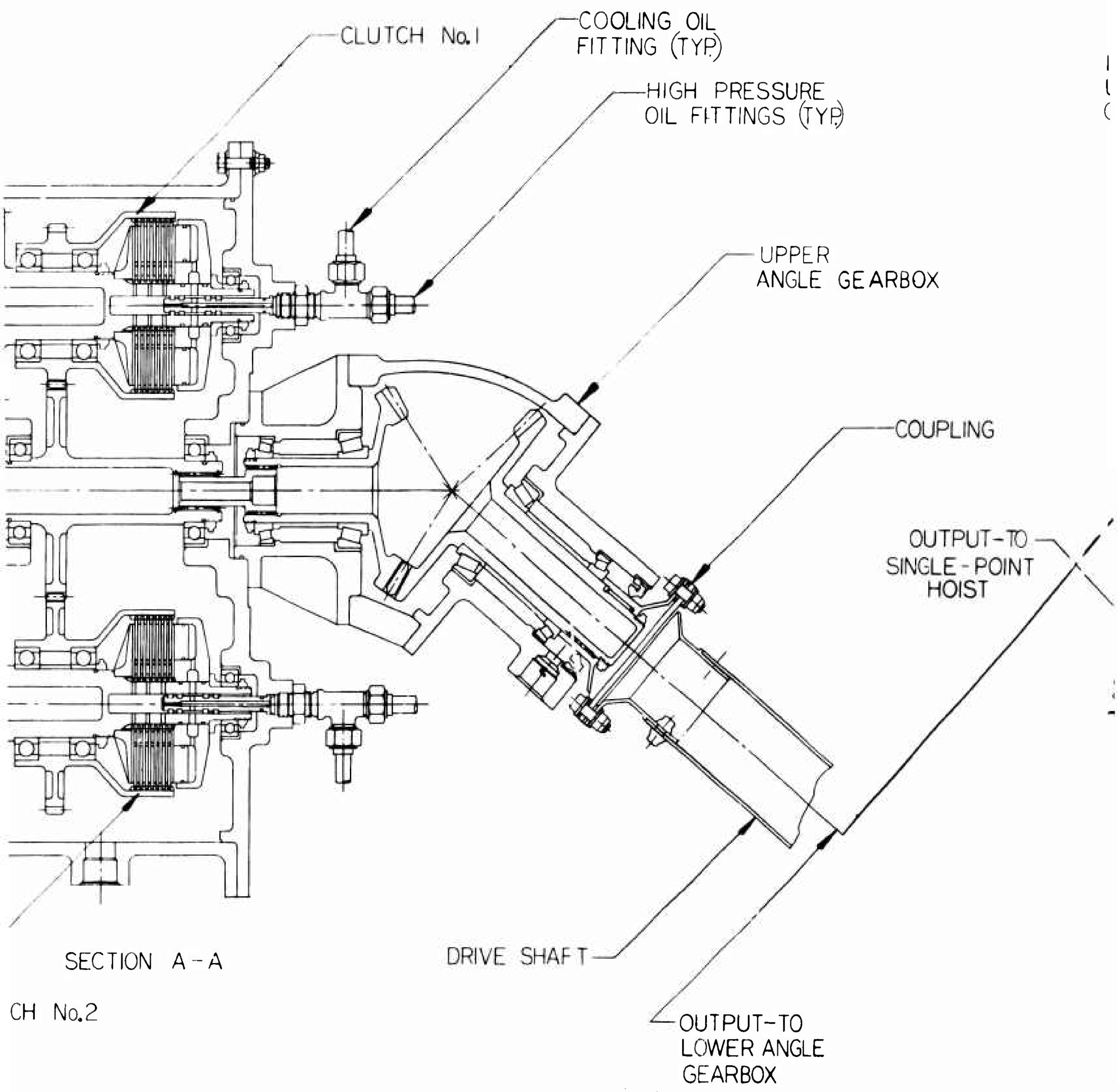


Figure 72. Clutch-Reverser Unit and Angle Gearboxes.

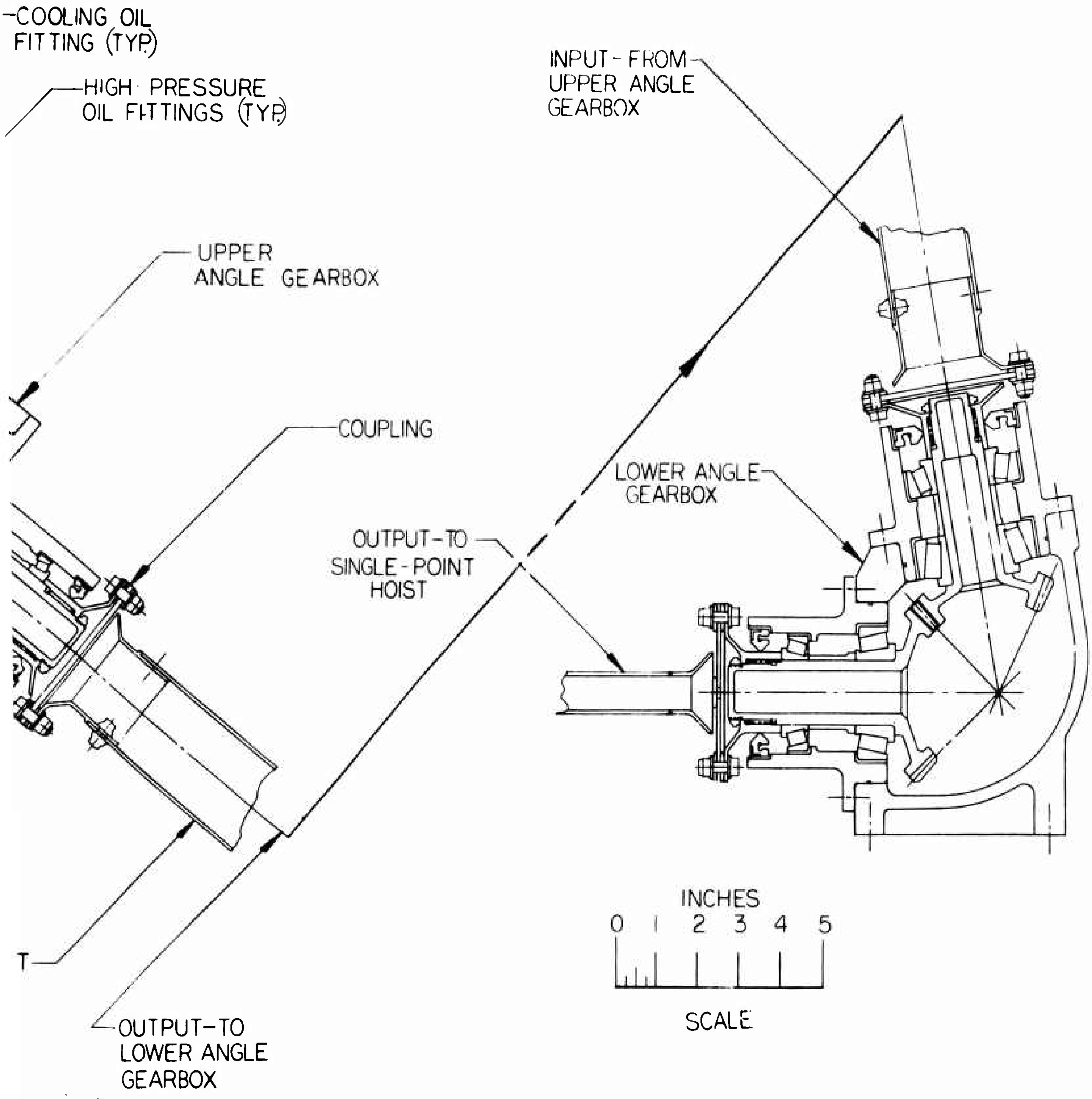
A.



SECTION A-A

CH No.2

B.



-COOLING OIL
FITTING (TYP)
HIGH PRESSURE
OIL FITTINGS (TYP)
INPUT - FROM
UPPER ANGLE
GEARBOX
UPPER
ANGLE GEARBOX
COUPLING
LOWER ANGLE
GEARBOX
OUTPUT - TO
SINGLE - POINT
HOIST
T
OUTPUT - TO
LOWER ANGLE
GEARBOX
INCHES
0 1 2 3 4 5
SCALE

C,

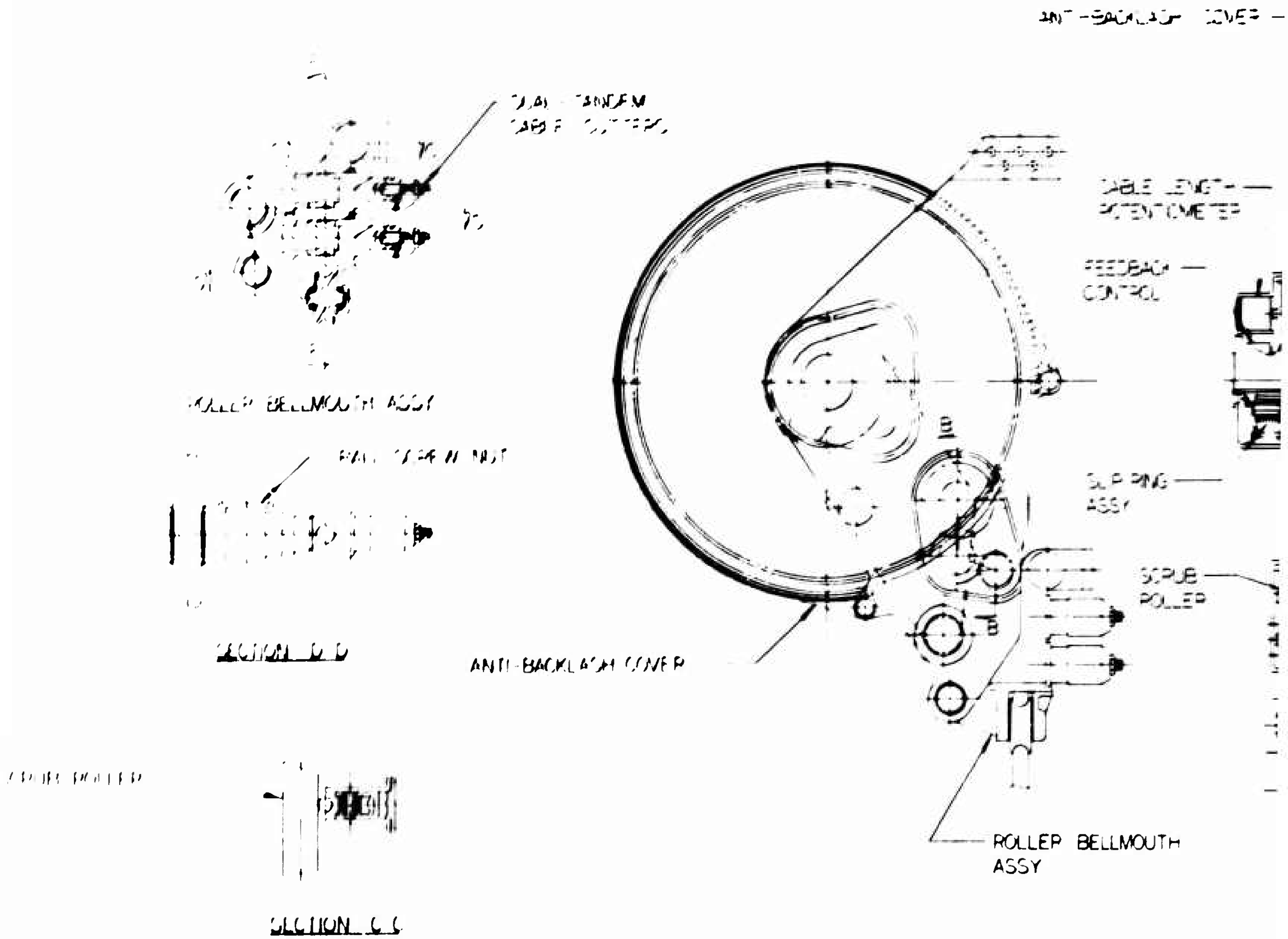
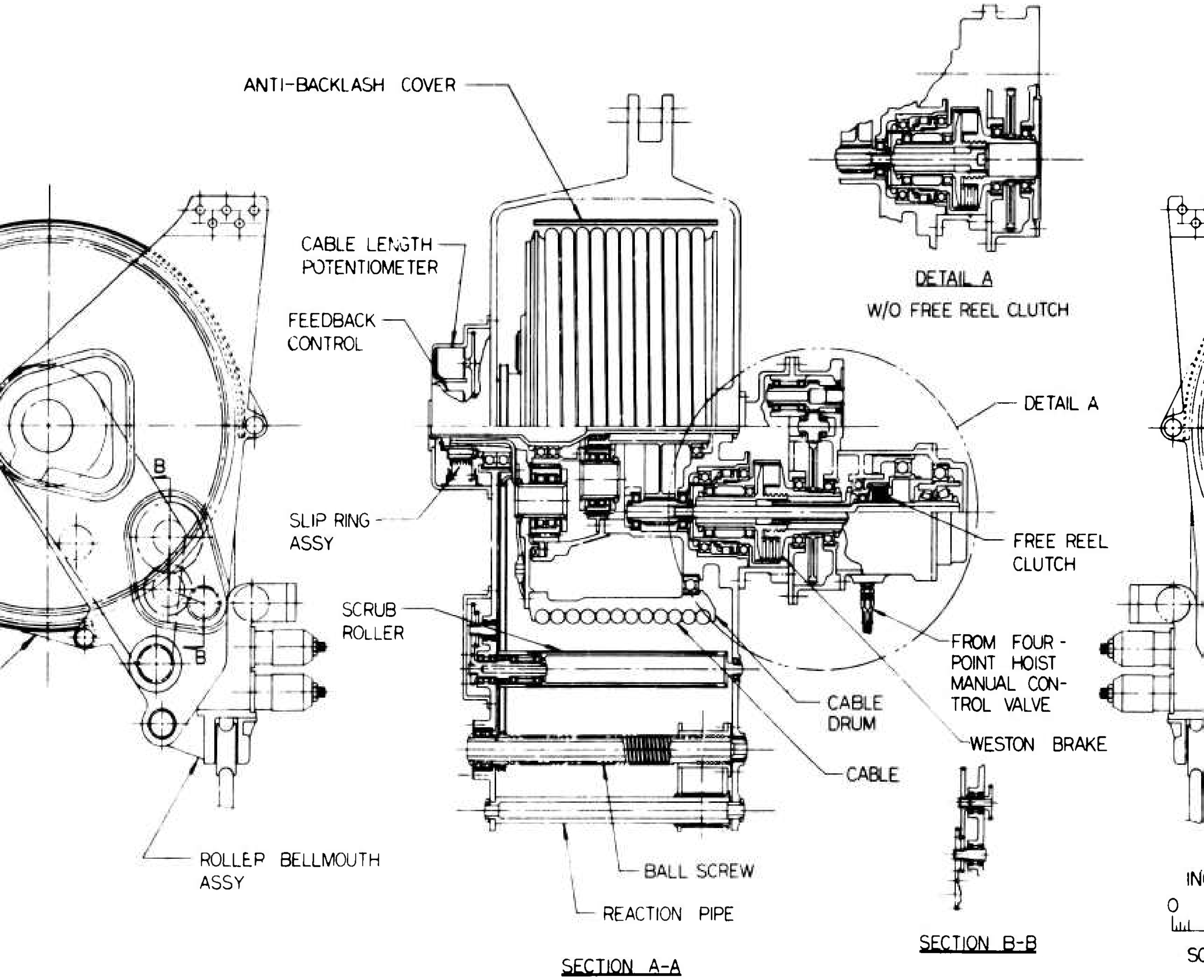
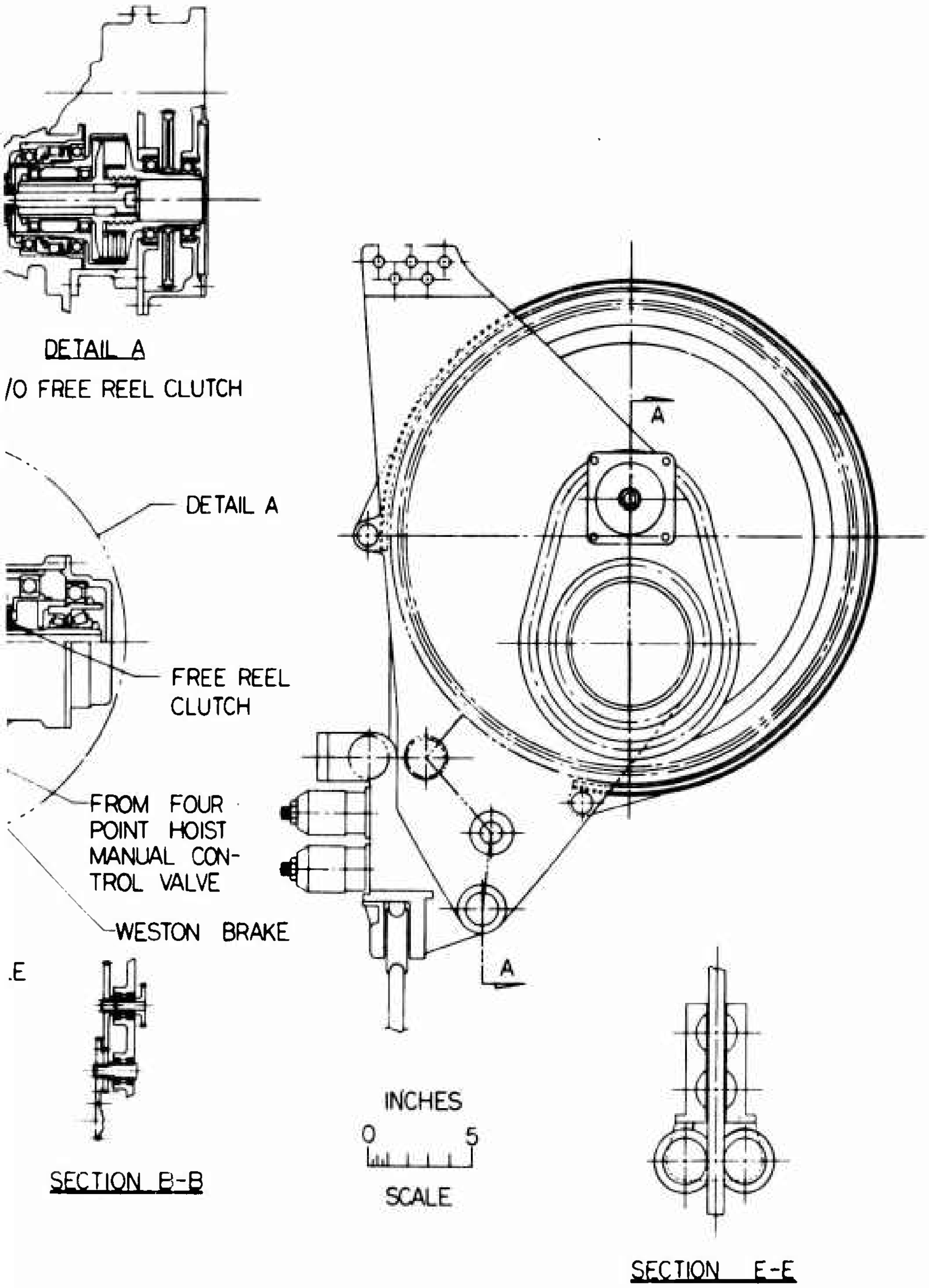


Figure 73. Four-Point Hoist.

A.


ANTI-BACKLASH COVER
CABLE LENGTH POTENTIOMETER
FEEDBACK CONTROL
SLIP RING ASSY
SCRUB ROLLER
B
B
ROLLER BELLMOUTH ASSY
BALL SCREW
REACTION PIPE
SECTION A-A
DETAIL A
W/O FREE REEL CLUTCH
DETAIL A
FREE REEL CLUTCH
FROM FOUR-POINT HOIST MANUAL CON-TROL VALVE
CABLE DRUM
WESTON BRAKE
CABLE
SECTION B-B

B.


DETAIL A
/O FREE REEL CLUTCH
DETAIL A
A
FREE REEL
CLUTCH
FROM FOUR
POINT HOIST
MANUAL CON-
TROL VALVE
WESTON BRAKE
.E
A
SECTION B-B
INCHES
0
5
SCALE
SECTION E-E

C1

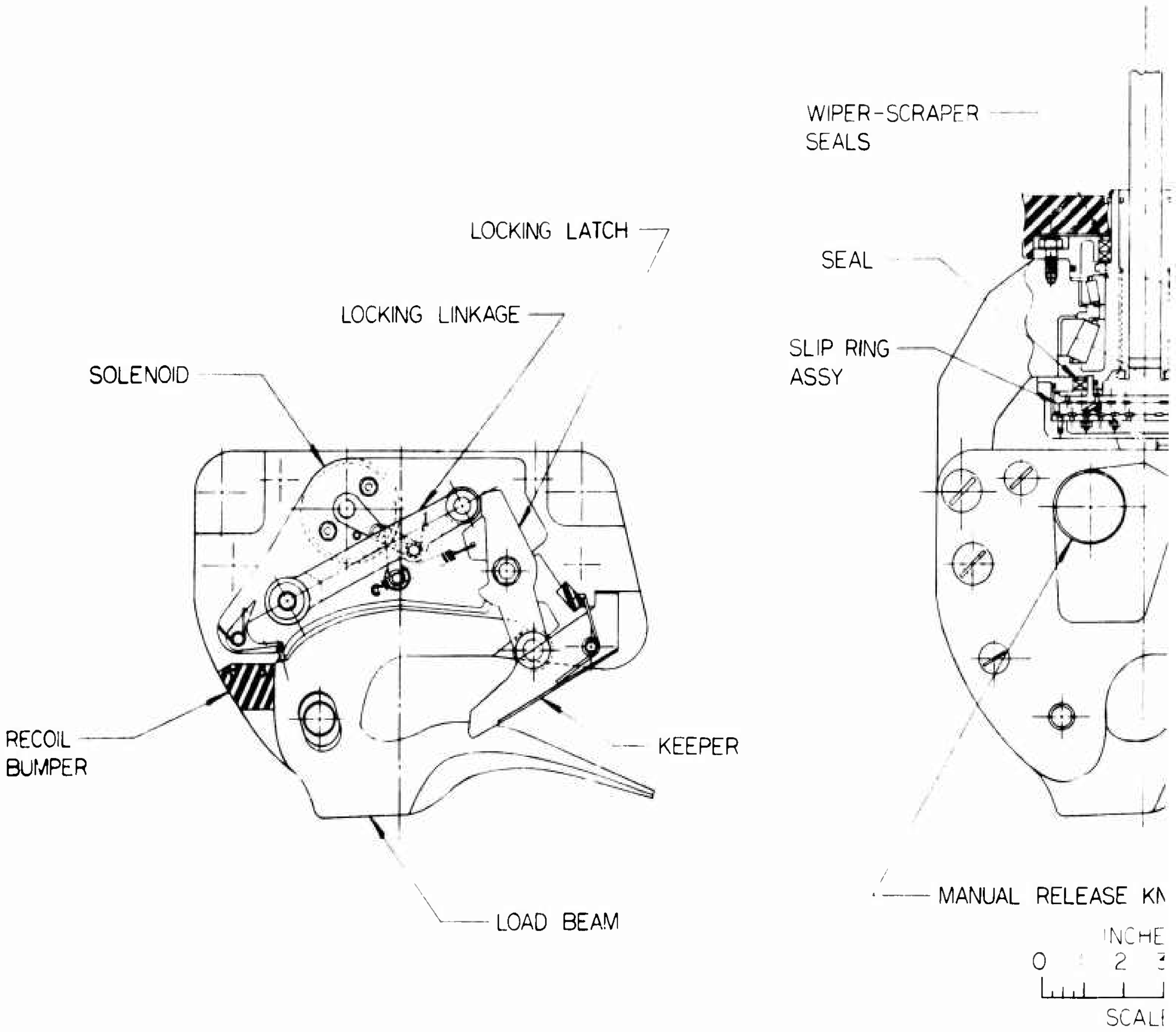
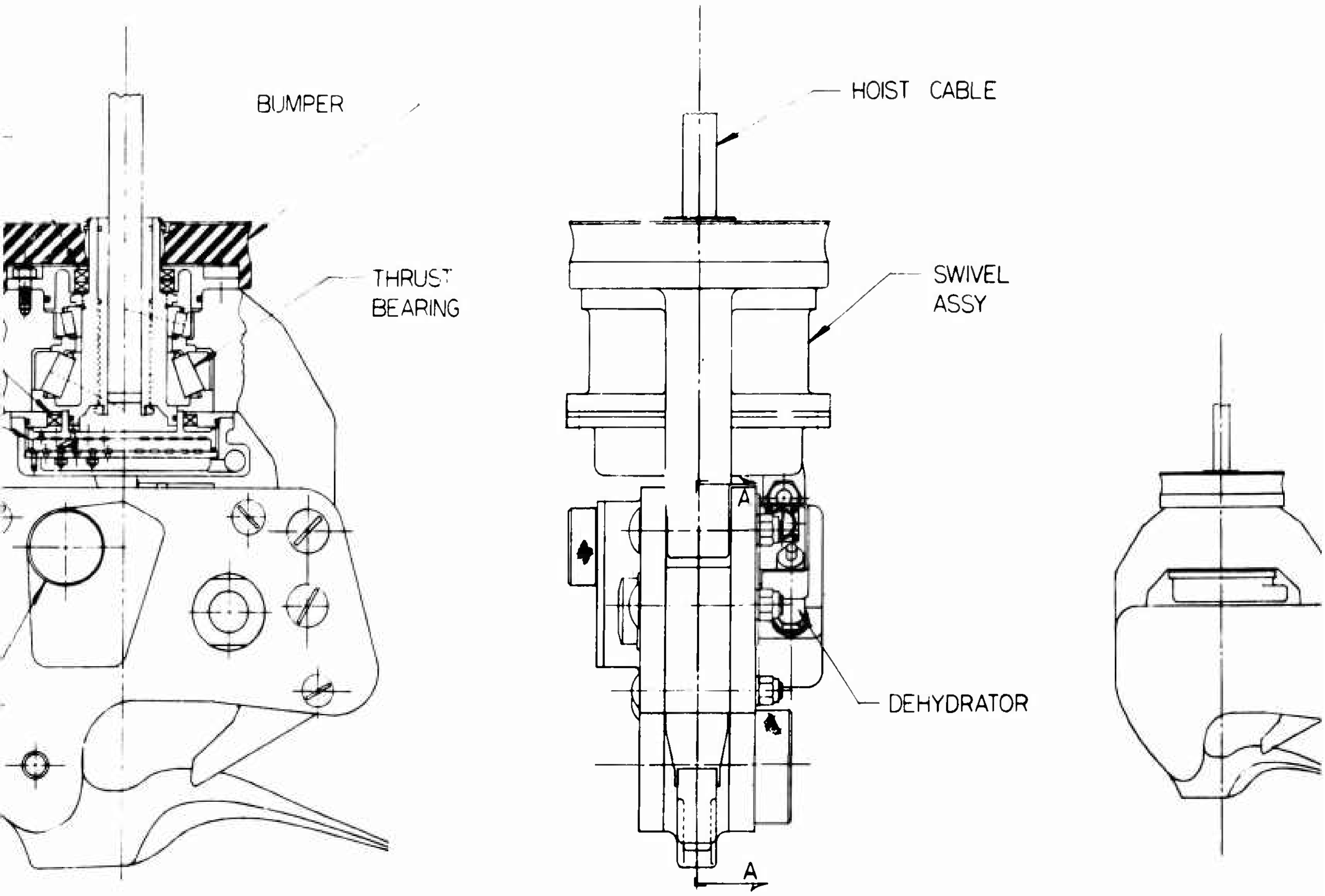


Figure 74. Cargo Hook, 12,000-Pound Capacity.

A.

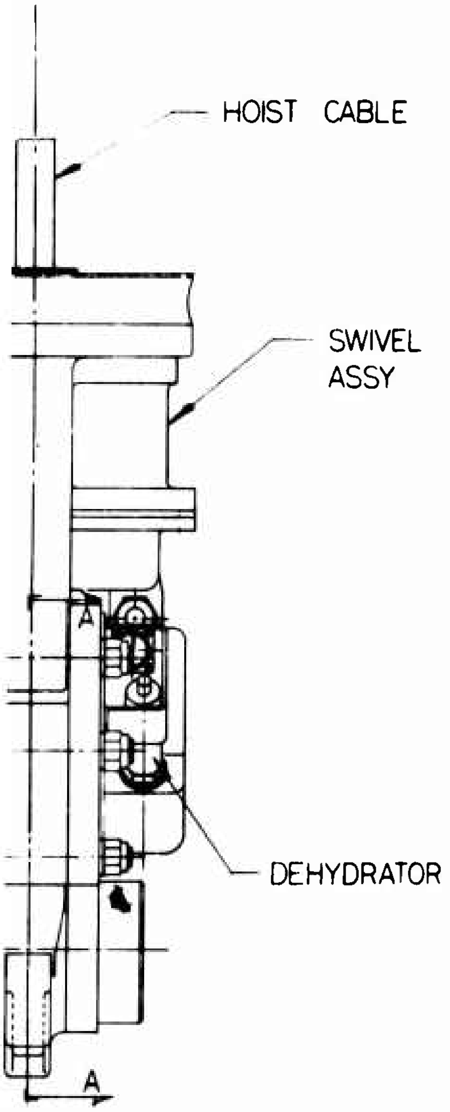


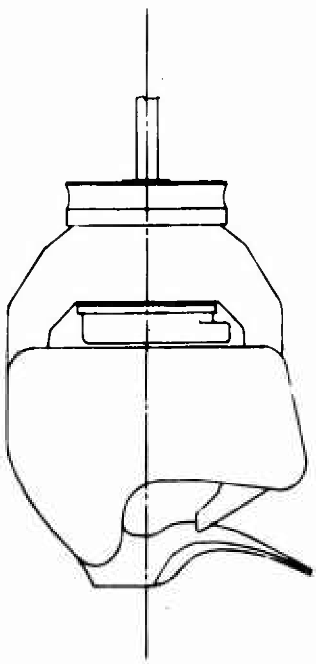
RELEASE KNOB

INCHES
0 1 2 3 4 5
SCALE

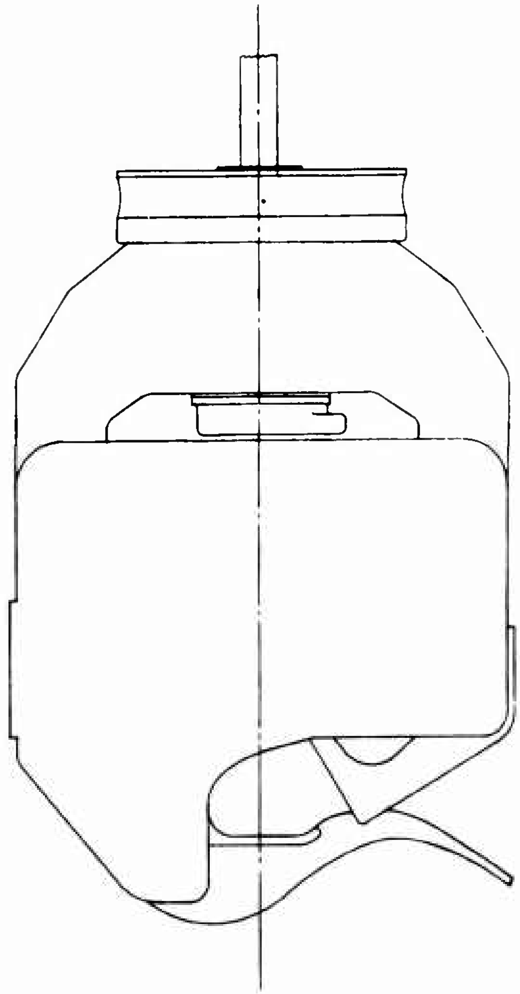
11,550-LB-CAPACIT
HOOK-SWIVEL ASS'Y

B.





11,550-LB-CAPACITY
HOOK-SWIVEL ASSY



40,000-LB-CAPACITY
HOOK-SWIVEL ASSY

SIZE COMPARISON

INCHES
0 1 2 3 4 5
SCALE

C.

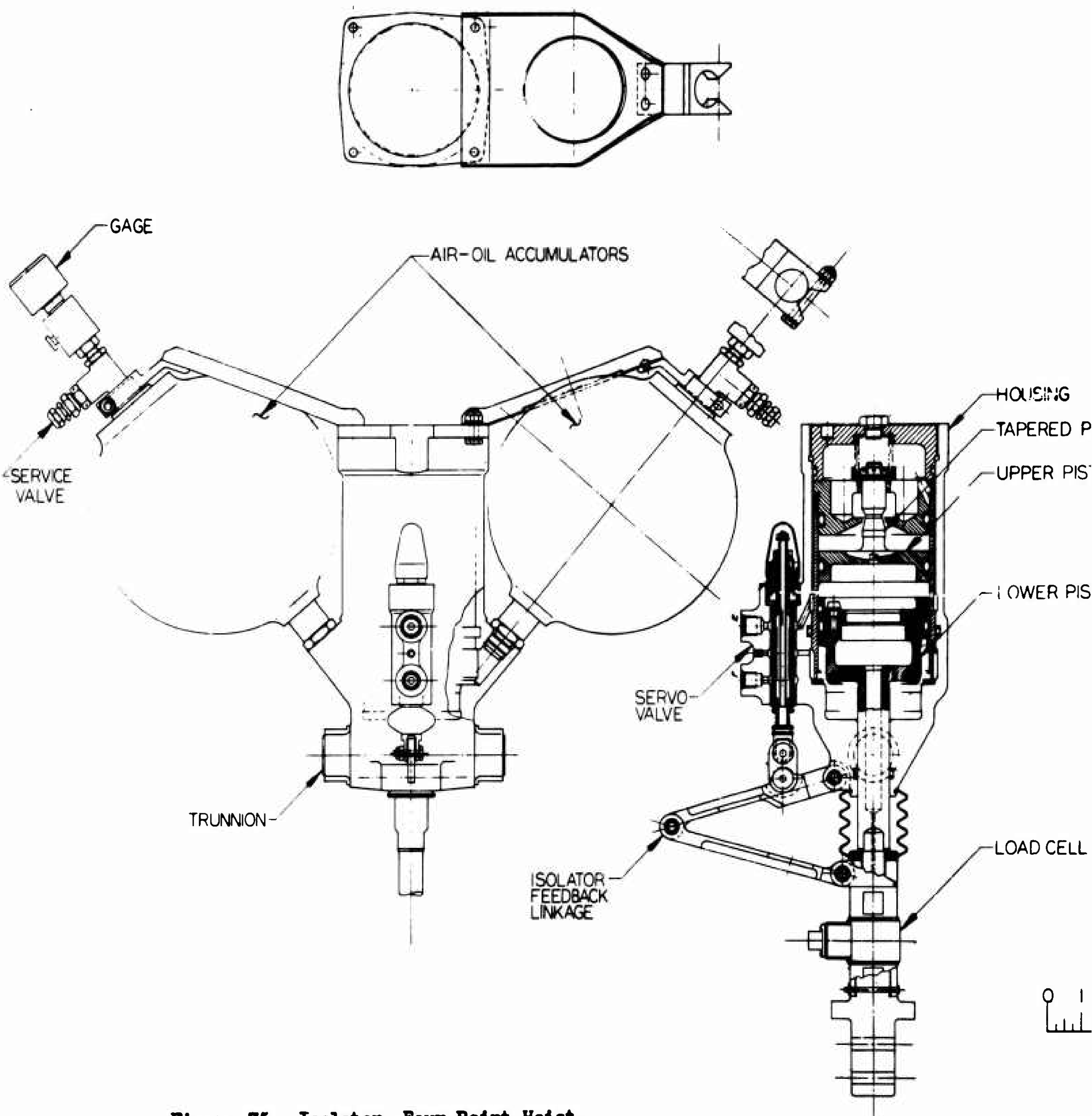
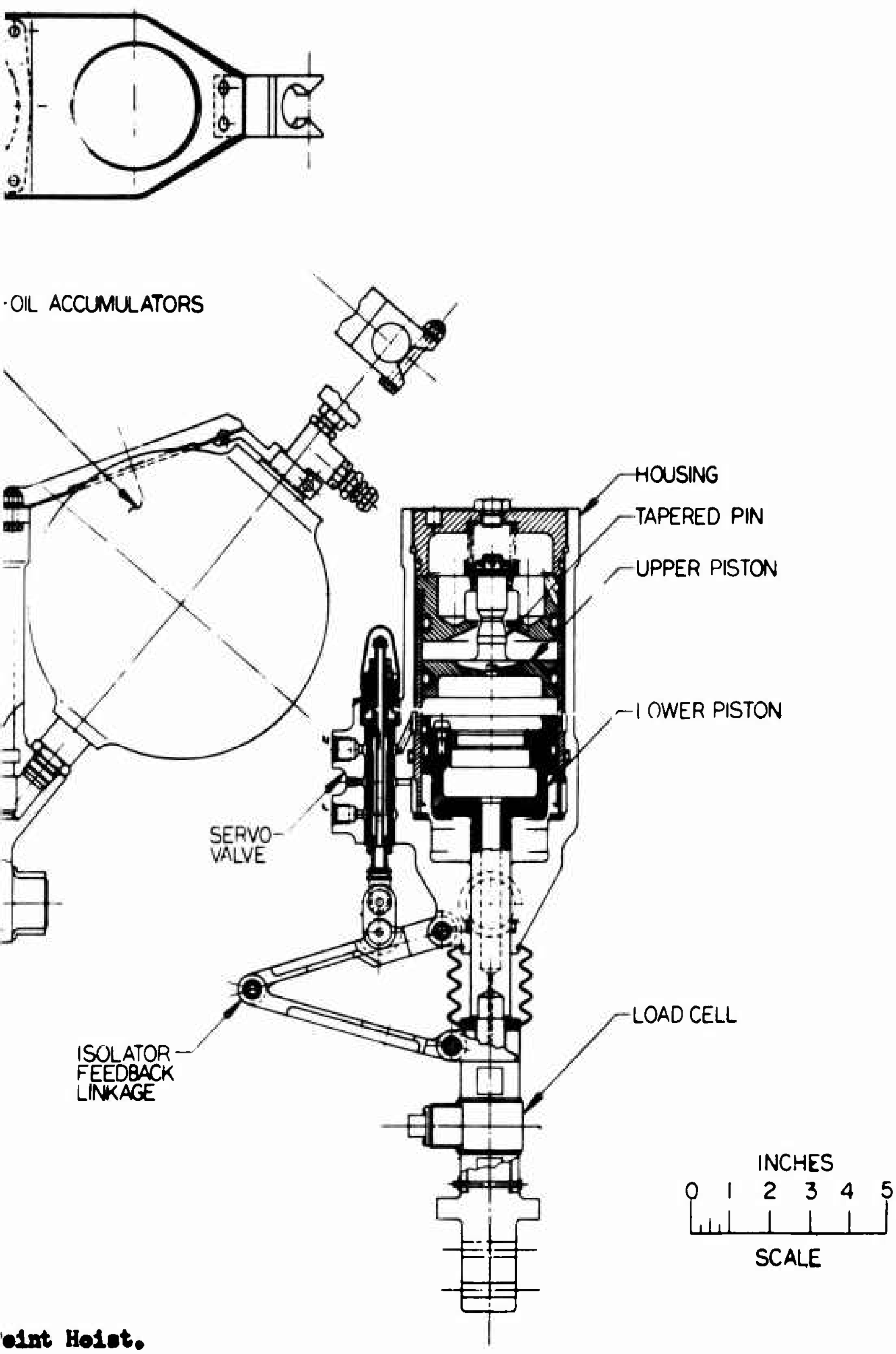


Figure 75. Isolator, Four-Point Hoist.

A.



OIL ACCUMULATORS
HOUSING
TAPERED PIN
UPPER PISTON
LOWER PISTON
SERVO-VALVE
LOAD CELL
ISOLATOR FEEDBACK LINKAGE
INCHES
0 1 2 3 4 5
SCALE

...oint Hoist.

B

APPENDIX IV
TYPICAL SEQUENCE OF OPERATIONS

INTRODUCTION

To provide a better understanding of the operation of the proposed single- and four-point cargo handling systems, typical missions involving both systems are described below. Two missions are described for the four-point system. The second mission is included to show methods that can be used to assure safe operation under adverse loading conditions.

No in-flight emergencies are described. Standard procedures for in-flight emergencies for the single-point mission, in all cases, should be the jettisoning of the load by the hook release method. If a malfunction did not permit the hook to release the load, the cable would then be sheared by the tandem-dual cutters or would be free reeled off the drum. In a multi-point mission no hook release would be attempted. Instead, the cables would be sheared by using the tandem-dual cable cutters.

Prior to all missions the cargo handling system should be checked out by the crew chief, with the APP providing the power, before the pilots enter the aircraft.

SINGLE-POINT MISSION

Mission: Fly to pickup area for bulldozer which is to be transported to clear area for observation post.

Terrain: Bulldozer in level field but 70-foot trees surround drop area on mountain top.

Load: Prerigged in a sling with sling legs attached to a nylon ring at apex.

Sequence:

1. Aircraft flown to pickup area, hover over bulldozer.
2. Reel out cable, ground crew slides nylon ring on load beam of hook.
3. Load is lifted off the ground by the aircraft.
4. Pilot checks aircraft controllability. If satisfactory, he signals aft pilot (hoist operator) to reel in on single-point hoist to cable length required for best flying qualities.
5. Aircraft is flown to drop site and hovers above trees.
6. Hoist is reeled out and hook is placed in auto touch-down mode.
7. When load is placed on the ground and the cable tension drops to 150 pounds, the hook opens, releasing the load.

8. Hook control is placed in safe and hoist cable is reeled in.
9. Aircraft departs drop site when hook is reeled in sufficient distance to assure clearance with tail rotor.
10. Aircraft returns to base.

Note: In the event of a malfunction of the automatic touchdown release, the electrical release would be used.

FOUR-POINT MISSION - STANDARD LOAD

Mission: Fly to pickup area for self-propelled mortar which is to be transported to forward area.

Terrain: Vehicle located in level field, to be put down on relatively rough terrain.

Load: Rigged for four-point pickup; no single-point sling available.

Sequence:

1. Aircraft flown to pickup area and landed near vehicle.
2. Vehicle driven under aircraft; hookup is made by ground crew.
3. Hoists reeled in until load is a foot off the ground.
4. Cable load indicators are checked to ensure that load falls within the C.G. limits of the aircraft.
5. Aircraft is lifted off into a hover; flight controls checked out as satisfactory.
6. Aircraft flies to drop site, which is found to be too uneven to permit landing.
7. A low altitude hover is established.
8. Hoists are reeled out until vehicle is several feet below the wheels of the aircraft.
9. Hover altitude is slowly reduced until load is on the ground and all four cables are slack.
10. Electrical hook release is actuated and all four hooks open.
11. Aft pilot (hoist operator) confirms that all hooks have released and aircraft hovering altitude is slowly increased until it is confirmed that all hooks are free.
12. Hoists are reeled in until a safe length is reached.
13. Aircraft returns to base.

Notes: (1) Step 10 requires the use of electrical release of the hooks. If the auto touchdown release were provided and were to be used under these conditions, the load release could result in adverse loads being felt by the aircraft.

These loads would result if one side of the vehicle touched the ground first. This would cause the hooks on this side to open, and the resultant loss in load on the aircraft would cause it to roll about the hooks that had not released. For this reason, the automatic touchdown release is not provided for the four-point hoist hooks.

(2) If one or more hooks fail to open, a ground crewman must be available to climb up on load and manually release the hooks. If no crewman is available, or hook(s) cannot be opened, the hoist cable(s) can be free reeled off the drum(s) or sheared with the tandem-dual cable cutter(s).

FOUR-POINT MISSION - NONSTANDARD LOAD

Mission: Fly to pickup area for bulldozer which is to be transported to forward area.

Terrain: Vehicle located on rough terrain, to be put down on a road in forward area.

Load: Rigged for four-point pickup, but pickup points not symmetrically located about C.G. of vehicle.

Sequence:

1. Aircraft flown to pickup area.
2. Rough terrain and unknown condition of pickup points, or reasonable suspicion of same, results in a 15- to 20-foot hover being established over vehicle.
3. Hoists reeled out until hooks are on the ground and the cables are slack.
4. Hookup is made by ground crew.
5. Vehicle is slowly lifted off the ground by the aircraft; hoists are not reeled in.
6. Vehicle swings forward, as vehicle C.G. is too far forward relative to the pickup points.
7. Pilot corrects for load swing with azimuth control (cyclic control stick) but feels that too much forward stick is required to permit forward flight.
8. Pilot requests aft pilot (hoist operator) to trim load by reeling in on aft hoists.
9. Aft hoists are reeled in but vehicle assumes an extreme nose-down attitude (or maximum cable load is reached and hoists stall).

10. Hoist operator informs pilot that he has run out of trim control with the hoists.
11. Pilot rechecks cyclic control and decides that not enough improvement has been made to warrant an attempt at forward flight.
12. Pilot informs hoist operator that it is a "no go" and asks that load be leveled up.
13. Vehicle is leveled up by lowering aft hoist cables.
14. Hover altitude is slowly reduced until load is on ground and all four cables are slack.
15. Electrical release is actuated and all four hooks open.
16. If time permits, the pickup points on the vehicle are repositioned and another attempt is made or a sling is rigged to permit single-point lifting. A single-point sling, with adjustable length legs, could be quickly set to compensate for the nonsymmetrical C.G. of the load so that it could be carried level.

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13. ABSTRACT

This report presents the results of a two-phase feasibility and preliminary design study of load suspension configurations capable of meeting the external cargo handling system requirements of a 40,000-pound-payload heavy lift helicopter

In Phase I, Design Analysis, both separate function configurations (those that incorporate individual single- and multi-point hoists) and combined function configurations (multi-point hoists used to perform both single- and multi-point missions) have been investigated for both single- plus two-point and single- plus four-point load suspensions. This phase was primarily concerned with investigation of hoist types; methods of power transmission to the hoists; and selection of mechanical, hydraulic, and electrical components. A comparative evaluation of 13 system arrangements was made on the basis of weight, power, reliability, in-flight safety, versatility, and productivity.

The single- plus four-point system was determined to meet the requirements best and was recommended for the Phase II, Preliminary Design. This phase included the preparation of layout drawings, load and stress analysis of major components, a maintainability and reliability analysis, and the preparation of a component development plan. The single- plus four-point system weighs 4974 pounds for a hoist capacity of 40,000 pounds. The system has been designed such that the hoists of both systems are readily removable for missions requiring minimum empty weight. For single-point operation (four-point hoists removed), the system weighs 2738 pounds; for four-point missions (single-point hoist removed), 2704 pounds.

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